May-June, 1955

LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

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MEMBERS: Members shall be persons not less than 24 years of age who: (1) are engaged in research and instruction at technical schools, universities, and various publicly and privately supported institutions in the field of lubrication; or (2) have occupied recognized positions as lubrication engineers for a period of three or more (not necessarily consecutive) years prior to date of admission, involving the responsibility for or supervision of the development, selection, field use and application of lubricants as differentiated from other activities; or (3) are indirectly concerned with the field of lubrication, but possessing other qualifications of experience, knowledge, and accomplishment, and have manifested a particular interest in the purposes and welfare of the Society to the extent that their membership would be a valuable contribution to the successful functioning of its activities. Fee $12.50.

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LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Vol. 11, No. 3 **May-June, 1955**

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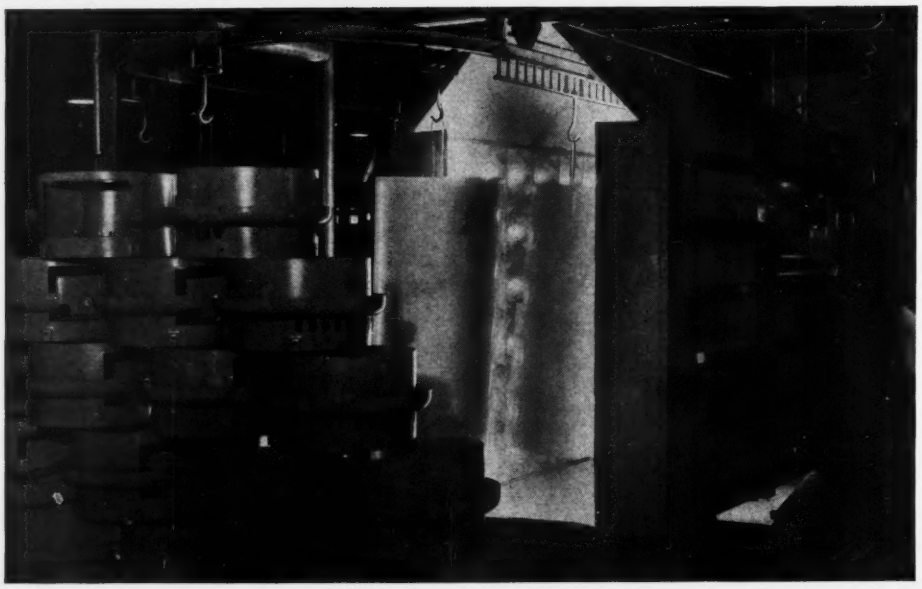

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Meet the Authors

Meet the Authors of the papers appearing in this issue of "Lubrication Engineering" (Vol. 11, No. 3, May-June, 1955). Mr. Archibald is the author of the series entitled 'Men of Lubrication;' Mr. Marshall served as Chairman of the Panel on 'Lubrication Problems in the Steel Industry.'

F. R. Archibald (p. 163)
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National Bureau of Standards

E. E. Bisson (p. 164)
NACA, Lewis Flight Propulsion Lab.

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F. Osterle (p. 187)
Carnegie Institute of Technology

J. A. Prevel (p. 178)
General Motors Corp.

E. Saibel (p.,187)
Carnegie Institute of Technology

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National Bureau of Standards

H. S. White (p. 182)
National Bureau of Standards

W. J. Wojtowicz (p. 174)
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J. F. Wygant (p. 193)
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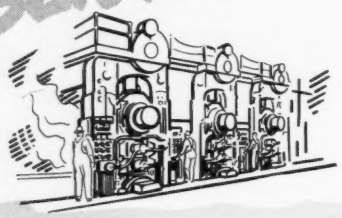

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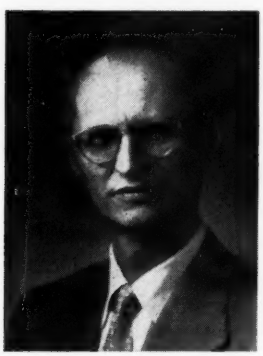
ANNUAL MEETING HI-LITES. The 10th Annual Meeting & Exhibit of the American Society of Lubrication Engineers, held April 13-14-15, 1955, at Hotel Sherman, Chicago, Illinois, attracted the largest attendance in the history of the Society to date: **1,407 registrants.** Thirty-two exhibitors combined to make this the largest exhibit ever held; a traffic count established the fact that 3,480 persons entered the Exhibit Hall in this three-day period. Fifty-eight papers were presented during the sixteen sessions, and 98 registrants attended the Lubrication Engineering Course sponsored by the ASLE Education Committee. George Spehn of D. A. Stuart Oil Co. acted as Toastmaster at the ASLE Chicago Section Welcome Luncheon, with guest speaker William E. Downes, Jr., a referee of a national football league, speaking on "An Engineering Approach to Professional Football." Over 450 persons attended the Annual Banquet (making it, too, the largest in Society history), with Prof. Burgess Jennings of Northwestern University acting as Toastmaster, and featuring Dr. Hubert N. Alyea of Princeton University as guest speaker (subject: "Atomic Energy, Weapon for Peace"). Following the introduction of ASLE Officers & Directors for 1955-56, Dr. E. R. Booser, General Electric Co., announced the award winners, with Retiring President John Boyd, Westinghouse Research Laboratories, presenting the awards as follows:



F. W. Ocvirk (pictured left), Cornell University, was awarded the **Alfred E. Hunt Memorial Medal,** sponsored by the Aluminum Company of America, for the best paper on the subject of lubrication in 1954 entitled "Measured Oil Film Pressure Distribution In Misalined Plain Bearings" **(Lubrication Engineering,** Vol. 10, No. 5, p. 262).

H. N. Kaufman (pictured right), Westinghouse Research Laboratories, was presented the **Walter D. Hodson Award,** awarded annually for the best paper on lubrication or an allied subject by a member 30 years of age or less, for his paper entitled "Classification Of Bearing Damage" ("Interpreting Service Damage In Rolling Type Bearings," p. 1, an ASLE publication).

F. P. Bowden, Cambridge University (Gonville & Caius College), England, received the **ASLE National Award,** constituting honorary life membership in the Society, in recognition of his outstanding scientific achievement in the field of lubrication and for his contributions to the science of lubrication in the United States and England.

H. O. Walp, SKF Industries, was presented a **Citation** for the valuable contribution he has made to the Society and its members in his paper entitled "Recognition Of The Causes Of Bearing Damage As An Aid In Prevention" ("Interpreting Service Damage In Rolling Type Bearings," p. 8, an ASLE publication).

PRACTICAL INFORMATION PLEASE! Every lubrication engineer counts as one of his valuable assets a fund of information accumulated from his experience. Since not every engineer has the same experience, the mutual advantages of sharing should be obvious. Large or small items of practical interest, when recorded, help to build up the body of knowledge which the lubrication engineer can draw on when necessary. The total information potentially available from this source is enormous.

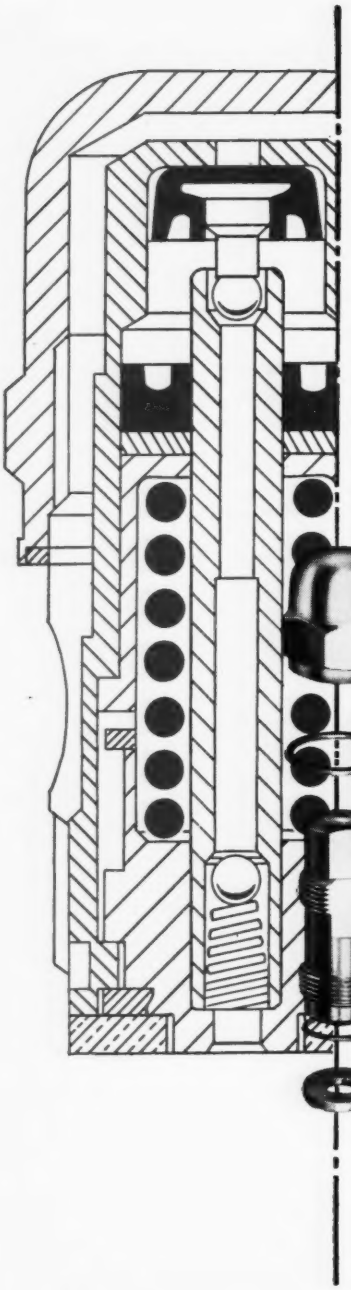
However, most engineers hesitate to impart such information, partly out of inertia, and partly from the feeling that their story does not fit into the category of a formal paper. To fit this situation **Lubrication Engineering** has instituted the "Experiment & Experience" department, expressly for the purpose of publishing short articles or stories regarding specific points in the practice of Lubrication Engineering. It especially welcomes descriptions of new departures from older practice, short experiments and results, specific solutions to current problems, even if not of general applicability, individual points of view of lubrication engineering practice, all the down-to-earth facts that can be published, one or more at a time.

The complaint is made (probably justifiably) that "Lubrication Engineering" does not publish enough material of a practical nature. No engineering journal can supply all of the answers to all the individual problems, but it can supply such of those as the possessors of the facts are willing to impart. The "Experiment & Experience" department offers both a challenge and an opportunity for lubrication engineers to gain practical information by sharing it. Put your contribution in the fund and the chances are that you will receive a larger one in return. (H. E. Mahncke)

CONVENTION POSTSCRIPTS. An organization can be said to have become adult when its major concerns turn from a preoccupation with how to get things done to actually doing them. The recent convention in Chicago seems to have marked such a point in the life of ASLE. There was an atmosphere in the committee meetings of "let's get on

(Continued on p. 198)

calls lubrication shots for hidden bearings!



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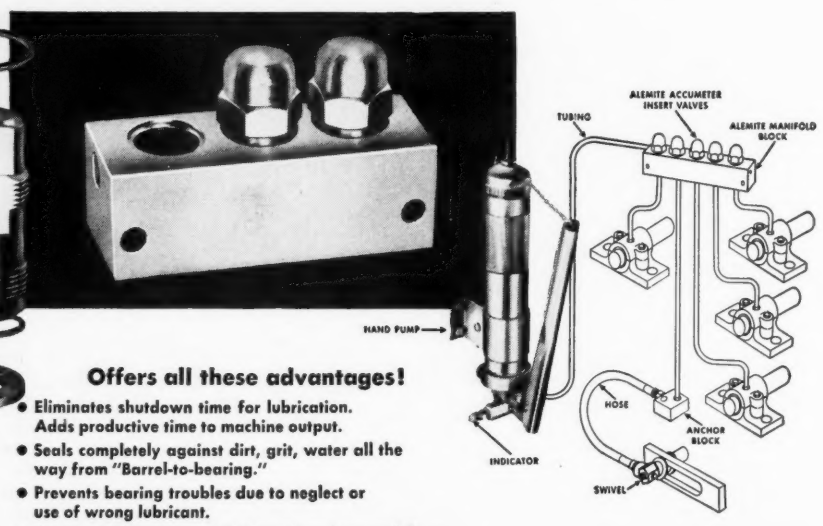

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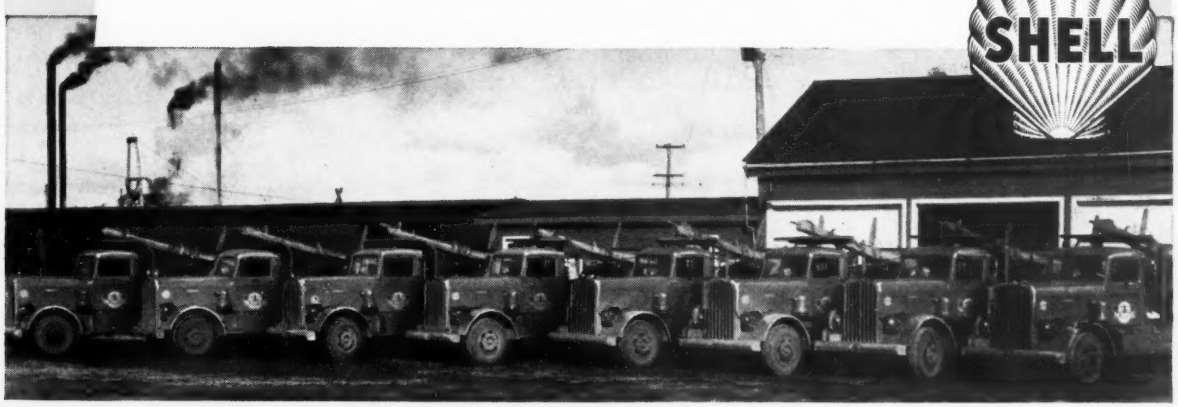

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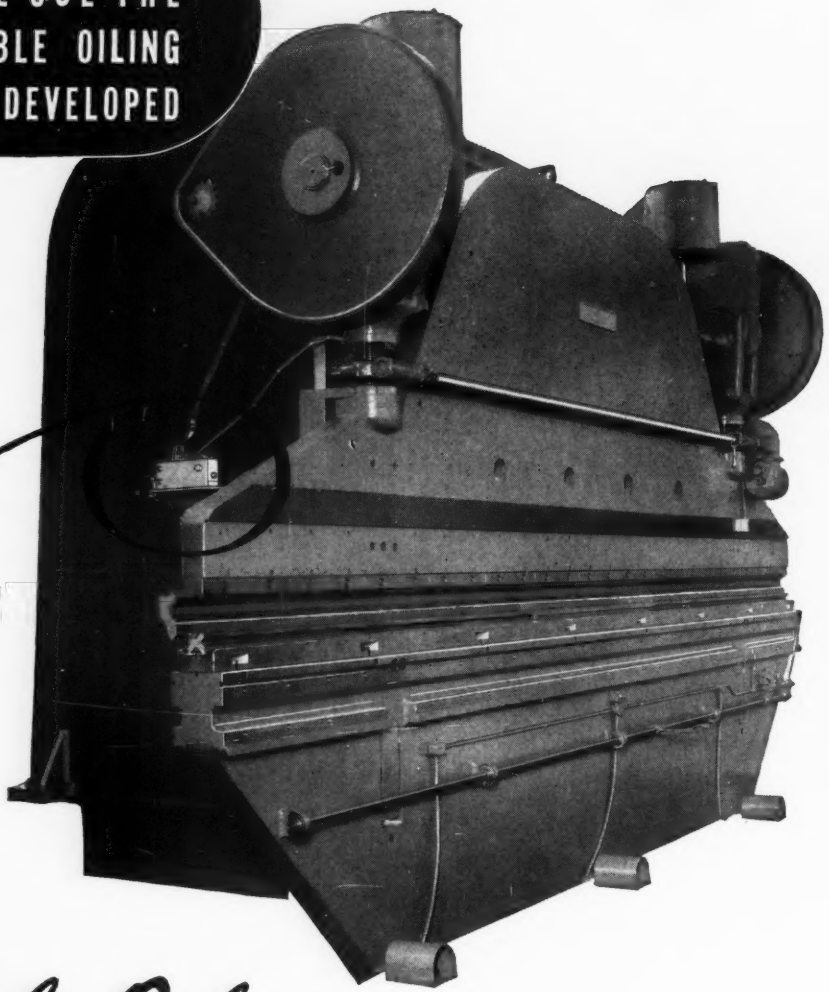

Illustrated is Madison-Kipp Lubricator Model FD installed as original equipment on a ⅜" by 20' Cincinnati Press Brake, manufactured by the Cincinnati Shaper Co., Cincinnati, Ohio.

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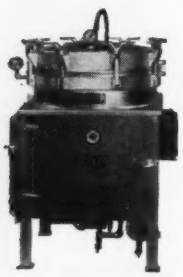

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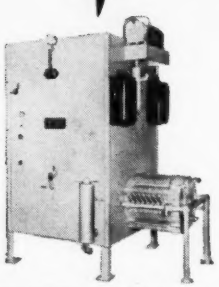

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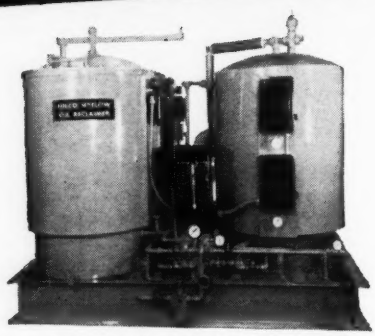

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Grease made with Bentone* 34 can be used on practically every type of machine and industrial equipment – and under almost every temperature and weather condition. Results are amazing: Grease consumption is less. Clean-up time is reduced. Savings realized by changing to Bentone grease have amounted to thousands of dollars annually.

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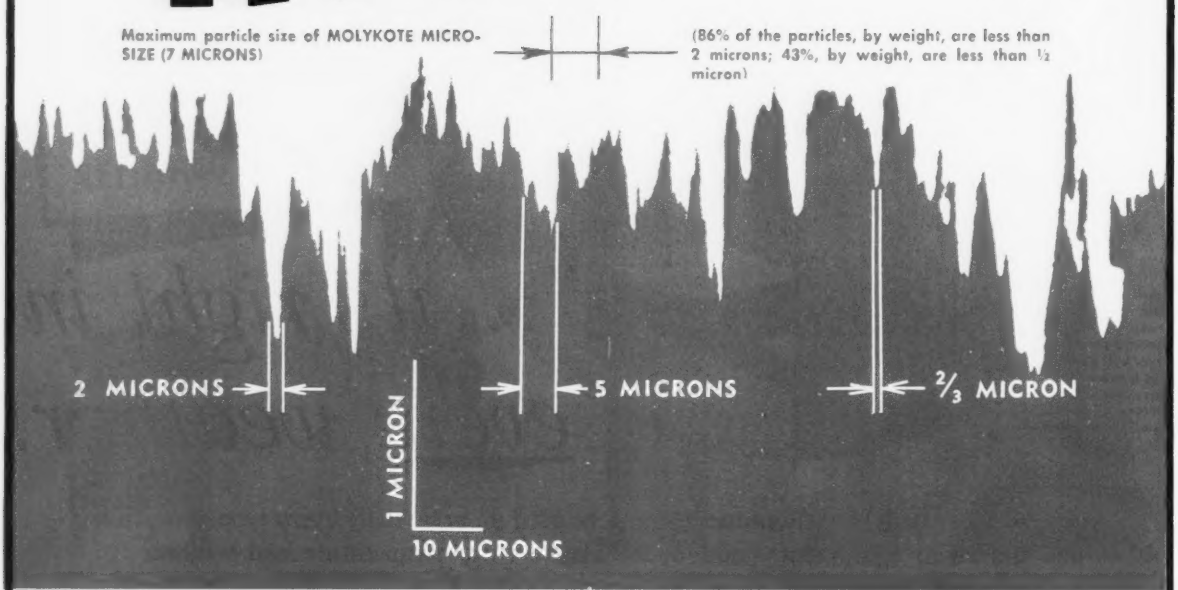
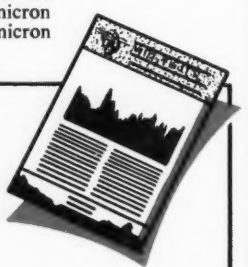

Maximum particle size of MOLYKOTE MICRO-SIZE (7 MICRONS)
(86% of the particles, by weight, are less than 2 microns; 43%, by weight, are less than ½ micron)
2 MICRONS
5 MICRONS
⅔ MICRON
1 MICRON
10 MICRONS

With average particle sizes of less than one micron MOLYKOTE MICROSIZE has the physical ability to enter the smallest spaces between mating parts to fully exploit the basic lubricating features of MoS_2 at extreme bearing pressures and temperatures.

Economically, too, the covering ability of MOLYKOTE MICROSIZE is far superior to ordinary grades of molybdenum disulfide powder.

The lower portion of the illustration shows in magnified cross-section a ground steel surface (finish 8 micro-inches R.M.S.). Taper sectioning was employed for study purposes resulting in a vertical magnification 25 times the horizontal (reference: "Mechanical Wear" edited by John T. Burwell, Jr.). Horizontal magnification, 1000X; vertical magnification, 25000X. In the illustration 1 mm equals 1 micron horizontally; 25 mm equals 1 micron vertically.

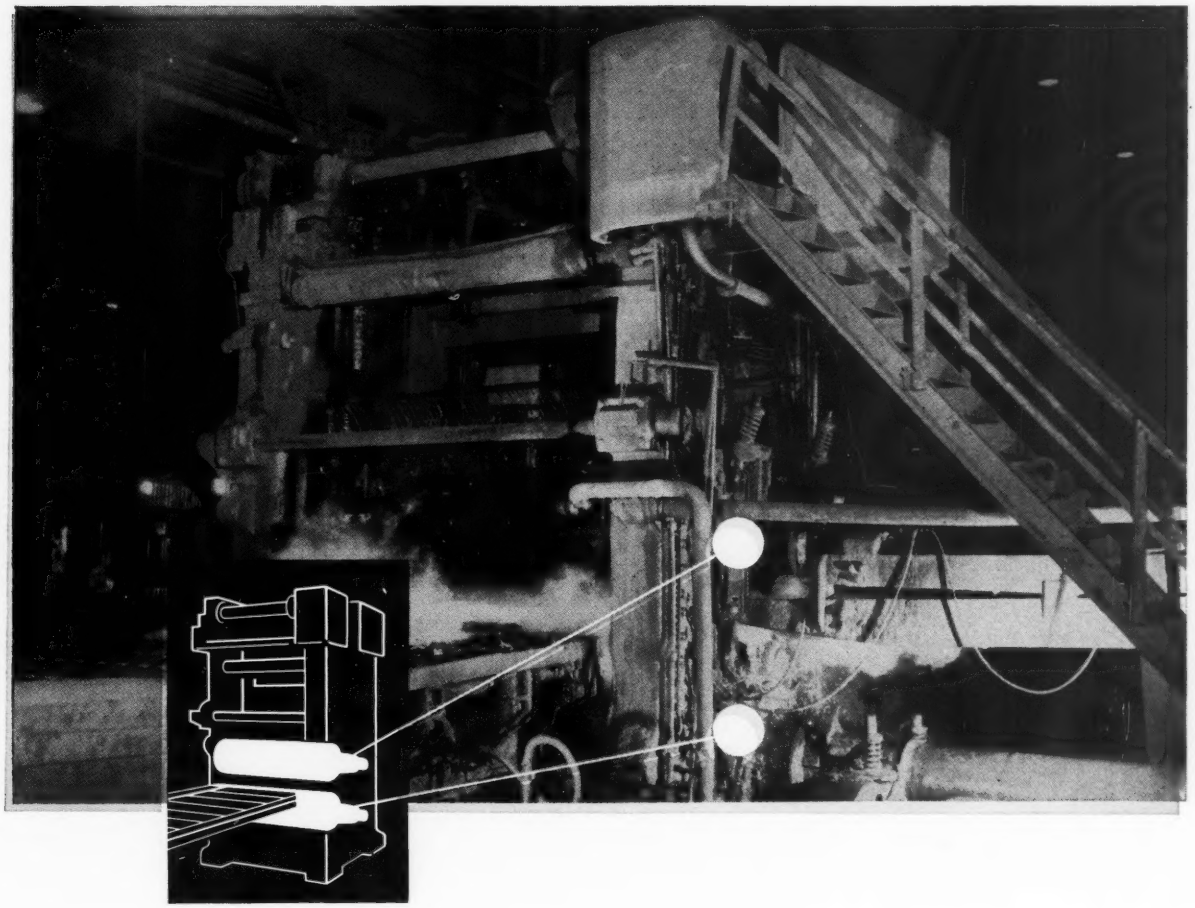


For complete details covering MOLYKOTE MICROSIZE, write today for Bulletin 104.
Note: Information on MOLYKOTE appears in Sweet's Product Design File 1955, Section 3C.

THE ALPHA CORPORATION

65 HARVARD AVENUE, STAMFORD, CONN.

Lubricating roll necks is no easy job



Lubricating the roll necks of a steel mill calls for a lubricant that will resist high temperatures. The lubricant must also function in the presence of steam, water, mill scale and other abrasive materials. It has to cushion great shock loads and cling tenaciously to the bearings it protects. It must last a long time . . . and it must be economical in use.

Atlantic Lubricant 10 performs exceptionally well in this service. It is being used in a pressure system on the mill illustrated, at the Lukens Steel Company in Coatesville, Pa., for the lubrication of roll neck bearings, pinion and spindle bearings.

Selecting the correct lubricant for the job is simplified when you call in your Atlantic Lubrication Engineer. He is a man with years of on-the-job knowledge of heavy industrial lubrication requirements of all kinds.

Why not see for yourself the value of Atlantic service? Just write, wire or phone the office nearest you for full information. The Atlantic Refining Company, Dept. L-6, 260 South Broad Street, Philadelphia 1, Pa.

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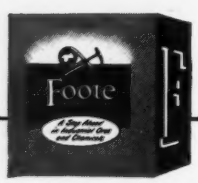
As a result of this ONE grease, inventory is greatly reduced and simplified lubrication procedures result in more efficient, economical operations.

It will pay you to contact your grease supplier for information about lithium base, multi-purpose lubricating grease.



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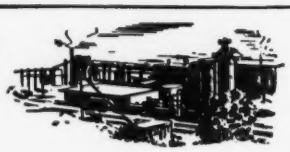

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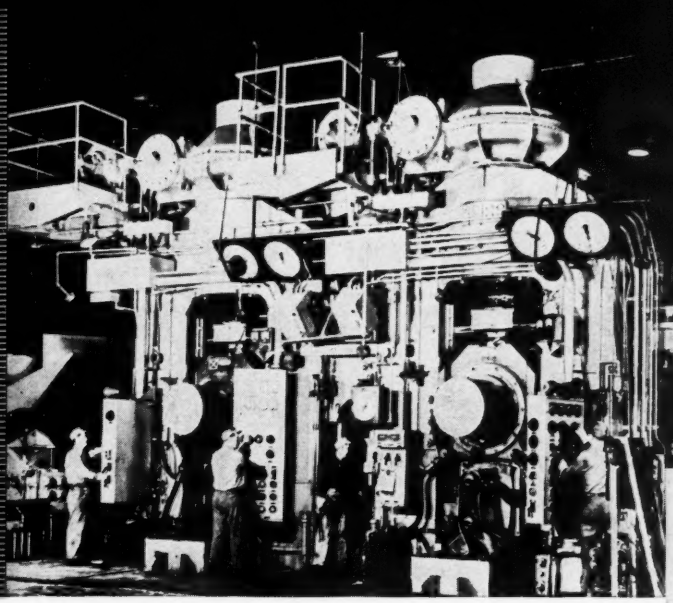

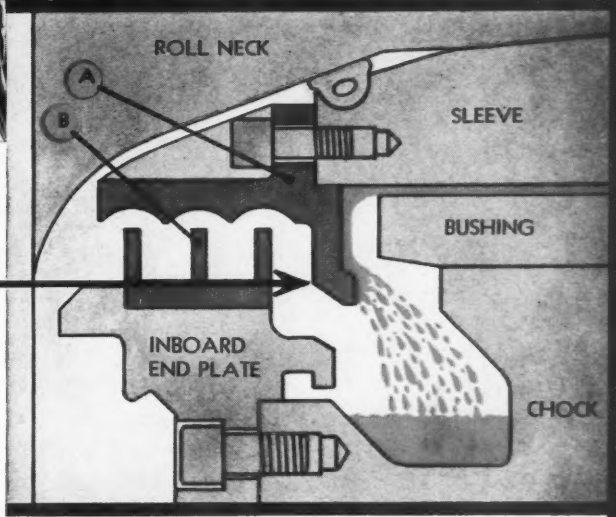

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A The rotating sleeve extension and flinger "A" throws oil from the bearing directly into the drain sump.

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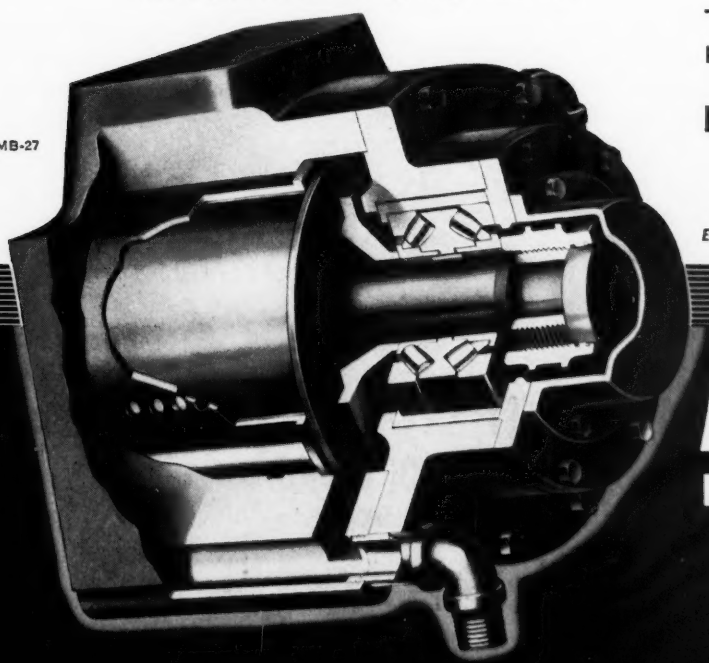

MB-27

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Coming Events

JUNE, 13-24, National Conference on Industrial Hydraulics, and Illinois Institute of Technology (Summer Course: "Basic Oil Hydraulics Power"), Technology Center, 3300 S. Federal St., Chicago, Ill.

19-23, American Rocket Society (Summer Meeting), Hotel Statler, Boston, Mass.

19-23, American Society of Mechanical Engineers (Diamond Jubilee Semi-Annual Meeting), Hotel Statler, Boston, Mass.

20-24, American Society for Engineering Education (Annual Meeting), Pennsylvania State Univ., State College, Pa.

21-24, Institute of Aeronautical Sciences (Annual Summer Meeting), IAS Bldg., Los Angeles, Calif.

27-29, American Society of Heating & Ventilating Engineers (Semi-Annual Meeting), St. Francis Hotel, San Francisco, Calif.

27 thru July 1, American Society for Testing Materials (58th Annual Meeting), Chalfonte-Haddon Hotel, Atlantic City, N. J.

27 thru July 1, American Institute of Electrical Engineers (Summer General Meeting), New Ocean House Hotel, Swampscott, Mass.

27 thru July 8, Massachusetts Institute of Technology, Metals Processing Div. (Summer Course: "Wear Theory in Metal Cutting & Bearing Design"), M.I.T., Cambridge, Mass.

JULY, 12-14, 2nd Western Plant Maintenance & Engineering Show, Pan Pacific Auditorium, Los Angeles, Calif.

AUGUST, 15-17, Society of Automotive Engineers (Golden Anniversary West Coast Meeting), Hotel Multnomah, Portland, Ore.

SEPTEMBER, 12-16, Instrument Society of America (10th Annual Conference & Exhibit), Shrine Exposition Hall & Auditorium, Los Angeles, Calif.

26-27, Independent Oil Compounders Assoc. (8th Annual Meeting), Bismarck Hotel, Chicago, Ill.

OCTOBER, 10-12, ASLE-ASME 2nd Annual Lubrication Research Conference, Antlers Hotel, Indianapolis, Ind.

ASTM 58th Annual Meeting, June 27 thru July 1

Seven timely symposiums will highlight the 32 technical sessions scheduled for the 58th Annual Meeting of the American Society for Testing Materials to be held at Chalfonte-Haddon Hall, Atlantic City, N. J., June 27 through July 1.

Symposiums will be presented on impact testing, judgment of factors in soil testing, high purity water corrosion, atmospheric corrosion of non-ferrous metals, speed of testing, and metallic materials for service above 1600 F. Sessions on related subjects will cover soils, non-ferrous metals, steel, tests of concrete, fatigue, and effect of temperature.

M.I.T. Special Summer Session, June 27 thru July 8

The Metals Processing Division of the Mechanical Engineering Department at the Massachusetts Institute of Technology will present a special two-week summer course on "Wear Theory in Metal Cutting & Bearing Design" from June 27 through July 8.

The several types of wear that normally contribute to the gradual deterioration of surfaces will be considered, and new developments in the theories that have been devised to explain the several types of wear will be discussed. Lecture topics will include the structure and properties of solids and lubricants, techniques for use in wear studies, wear studies involving radioactive tracer methods, temperatures and wear characteristics of cutting tools and grinding wheels, and machinability and machining economics.

Full details and application blanks for this course may be obtained from the Summer Session Office, Room 7-103, M.I.T., Cambridge 39, Mass.

ASLE-ASME 2nd Annual Lubrication Conference, October 10-12

The American Society of Lubrication Engineers and the Lubrication Activity of the American Society of Mechanical Engineers will hold their 2nd Annual Lubrication Research Conference at the Antlers Hotel, Indianapolis, Ind., on October 10-11-12.

As in the 1954 Conference, there will be no conflicting sessions, and those attending will have the opportunity to hear the presentation of all papers. The practice of providing registrants with preprints of the papers to be presented will be continued.

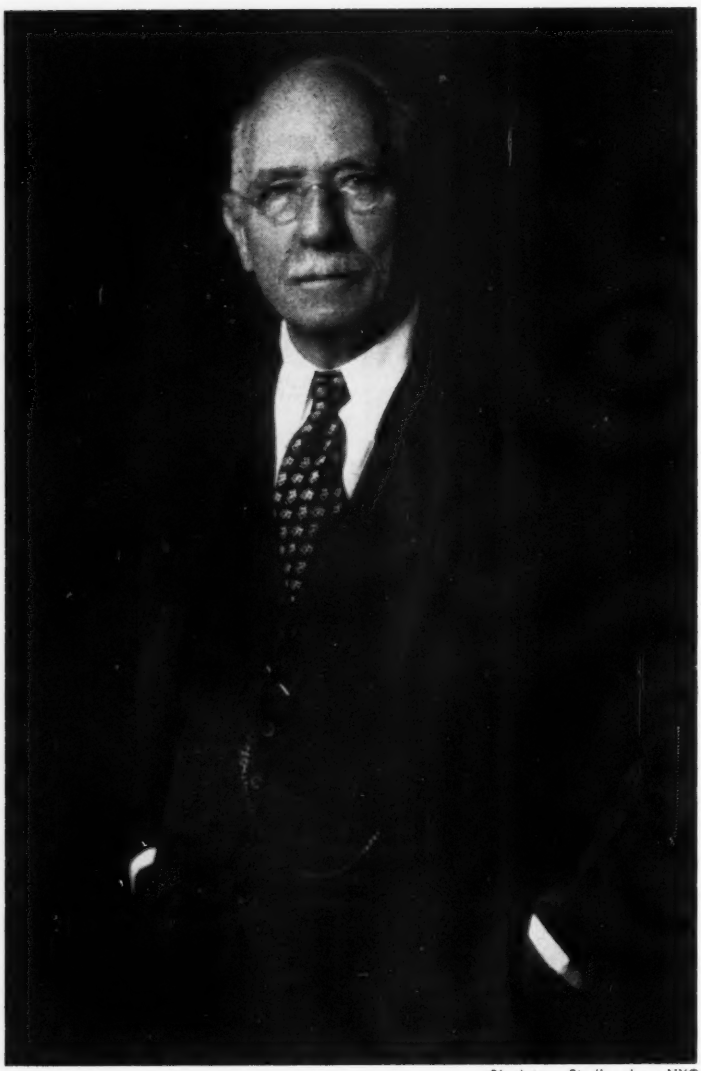
Based on the numbers of papers received, the Conference will be divided into the following sessions: (1) Bearing Instability, (2) Recent Studies in Hydrodynamic Lubrication, (3) Lubricants, (4) Rolling Contact Bearings, and (5) Boundary Lubrication.

(A more detailed program will appear in the August '55 issue of "Lubrication Engineering.")

ASLE 11th Annual Meeting & Exhibit, April 4-5-6, 1956

The American Society of Lubrication Engineers will hold their 11th Annual Meeting & Exhibit at the Hotel William Penn, Pittsburgh, Pa., on April 4-5-6, 1956.

Tentative technical sessions include: (1) Spray Application of Lubricants to Plane Roll Neck Bearings, (2) Steel Mill Cold Rolling Lubricants, (3) Boundary Lubrication, Friction & Wear, (4) Gear Lubrication, (5) Measurement of Properties of Fire Resistant Lubricants & Hydraulic Fluids, (6) Railroad Journal Bearings, (7) Plain Bearings, (8) Fire Resistant Fluids Today, (9) Metalworking Lubricants, (10) Rolling Contact Bearings, (11) Disposal of Waste Lubricants & Coolants with Reference to Stream Pollution, and (12) Rheology.



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ALBERT KINGSBURY

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Men of Lubrication

ALBERT KINGSBURY. The great industrial expansion of the United States during the twentieth century is intimately associated with the inventive genius of a relatively few men. One of the most scientific but least known of these men is Albert Kingsbury. His invention of the thrust bearing which bears his name was an engineering accomplishment of the first magnitude, though by the very nature of the intricate theory which lies behind its conception it has attracted little popular interest. But Kingsbury's thrust bearing has had a vital influence in power generation, in ship propulsion, and in numerous other departments of mechanical engineering which affect our daily lives in a very fundamental way.

Albert Kingsbury was born at Morris, Illinois, in 1863. He was descended from old New England families on both sides. His elementary education was obtained in Cuyahoga Falls, Ohio. After serving three years as a machinist's apprentice, he entered Ohio State University as a freshman in Mechanical Engineering in 1884 but dropped the course at the end of his sophomore year to work with one of the University Professors on a mechanical problem. This did not prove a very satisfactory arrangement and he found another job; this time as a machinist with the Warner & Swasey Company. In 1887 Kingsbury entered the junior class in Mechanical Engineering at Cornell University and graduated two years later when he was twenty-six years of age.

During his student period at Cornell, Kingsbury was for the first time brought in contact with lubrication problems. Dr. R. H. Thurston, the head of Sibley College, Cornell University, assigned Kingsbury the task of testing some bearing materials. These samples had already been tested and found to vary a great deal in the amount they had worn. From what followed, and because the matter is now much better understood, it must be concluded that the work had been done in a very poor way. Kingsbury started by scraping each bearing carefully to the shaft of the Thurston testing machine. This careful preparation of the bearing specimens produced in the test a condition of fluid film lubrication, and to the surprise of Dr. Thurston and Kingsbury no wear could be produced. The experience made a deep impression on Albert Kingsbury, but the explanation of his unexpected result was unknown to him until he read Osborne Reynolds' famous paper "On the Theory of Lubrication" about nine years later.

In 1889 Kingsbury accepted a teaching post at the New Hampshire College of Agriculture and the Mechanic Arts at Hanover, N. H. After one year of teaching he engaged in an unsuccessful business venture with an older cousin but, following this failure, returned to teaching again at the New Hampshire College; this time as a full professor of mechanical engineering. From then to the end of his life the subject of lubrication was the central theme of his career.

The anomalous experience with the bearing testing machine at Cornell was still much in his mind and he was soon engaged in various researches in lubrication. His work at that time on a torsion compression machine of his own construction was the basis of his paper of 1896, "Experiments on the Friction of Screws," presented before the American Society of Mechanical Engineers. In the construction of his torsion compression machine Kingsbury made a fortuitous but significant observation. The loading element of the machine was a 6 inch diameter hydraulic piston and cylinder. The piston was made to operate without packing and must have been a close fit for it was found that it would spin in the cylinder without lubrication. He concluded that the air was acting as the lubricant, and upon this theory proceeded to construct a model* provided with pressure taps to demonstrate the existence of the fluid pressure developed in the air film. This model aroused much interest in engineering circles of the time but its greatest value was to bring to Kingsbury's attention the great work of Osborne Reynolds. During one of the demonstrations of the model Kingsbury had made some statement not in accord with Reynolds' theory. Following the demonstration this was mentioned to him. It was Kingsbury's first knowledge of Reynolds' work and is, in retrospect, a dramatic point in his career.

Kingsbury immediately set to work on a study of Reynolds' paper and made more thorough experiments on his air lubricated model in the light of the hydrodynamical theory. His paper of 1897, "Experiments with an Air-Lubricated Bearing," (Journal of the American Society of Naval Engineers, Vol. 9, 1897), followed.

Osborne Reynolds had worked out the load capacity of an inclined flat plate moving over another flat plate of indefinite size, and this was now seen by Kingsbury to offer a solution to the very troublesome thrust bearing problem. In applying Reynolds' theory to the problem Kingsbury made two important contributions. Instead of producing a bearing in which one surface was constrained to move with an inclination to the other (a construction which would have been difficult to achieve), he proposed supporting one element of the bearing on a pivot located at the center of pressure. In this way the bearing shoe would take its proper inclination automatically. However, the theory showed that the center of pressure is not at the midpoint of the bearing, and hence a bearing constructed in the way he proposed would work for only one direction of rotation. After much thought he decided to proceed with a bearing having the pivots at the center of the shoes. This bearing was completely success-

(Continued on p. 197)

(This is the third in a series of articles by F. R. Archibald, Analyst, of Arthur D. Little, Inc.)

Effects Of Sliding Velocity & Temperature On Wear & Friction Of Several Materials

by R. L. Johnson, M. A. Swikert & E. E. Bisson*

Wear experiments were conducted with leaded brass, iron-silicon-bronze, cast Inconel and nodular irons sliding against hardened steel. Separate series of data show (1) the effect of various sliding velocities up to 18,000 feet per minute in room temperature runs, (2) the influence of ambient temperatures up to 1000 F. in runs made at low sliding velocity (120 ft/min) and (3) the influence of temperatures up to 650 F. in runs at a sliding velocity of 8000 feet per minute. Both sliding velocity and temperature had considerable effect on wear of unlubricated surfaces. The effects, however, were markedly different for the different metals. Changes in wear properties may have resulted from effect of temperature on physical properties and on the formation of oxide films. NiO films on cast Inconel are necessary in order to obtain low wear. Rate of wear in air decreased with increasing temperatures as oxides were more readily formed. A copper alloy had higher wear at 450 degrees when lubricated with a diester than when run without lubrication. Other materials did not show this tendency.

Wear or surface failure tendencies of materials for cages or retainers of rolling contact bearings have continued to be problems in the development of new turbine engines for aircraft.[1] High surface speeds and high operating temperatures make the cage material problem of bearings for turbine engine applications more critical than most other bearing applications. In particular, the high temperature requirement is becoming more severe as new engines are developed.[2, 3, 4, 5] It has been projected that bearing temperatures as high as 1000 F. merit consideration.[2] At high temperatures, materials with strength, oxidation resistance, dimensional stability, and hardness properties better than those of current materials are required. At present, bronzes are used for cages and SAE 52100 steel for races and rolling elements. New cage materials must be compatible (from a wear and surface failure standpoint) with the tool steels that are being considered for races and rolling elements of high temperature bearings. Compatibility is important because of inadequate lubrication between sliding surfaces of the cage and the races and rolling elements (Gurney, Ref. 1). Inadequate lubrication results in localized metal to metal contact.

Many experimental and several theoretical approaches have been suggested to explain the mechanism of wear and surface failure.[6 to 11] Under limited conditions of application several empirical expressions for wear can be shown to have merit. In general each of these wear expressions include several factors from an extensive list of conditions of operation and other metallurgical, chemical, physical, and mechanical variables. The cage material problem involves a large number of these variables over extreme ranges of conditions. These factors make the problem so complex that it appears impossible to advance any generally useful expression for wear of cage materials. Insofar as possible, however, all factors must be considered when planning and evaluating results from experimental wear studies. Many factors that are important in wear are also important in friction; this indicates that wear and friction problems cannot be separated. It

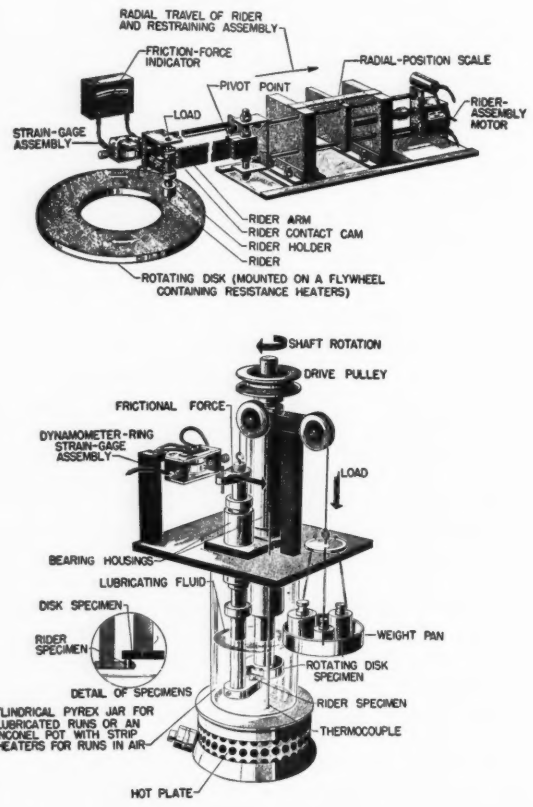

Fig. 1. Schematic diagrams of wear and friction equipment. (Top) Fig. 1(a), High sliding velocity apparatus; (Bottom) Fig. 1(b), Low sliding velocity apparatus.

*National Advisory Committee for Aeronautics, Lewis Flight Propulsion Laboratory, Cleveland, Ohio.

(This paper was co-sponsored by the ASLE Technical Committees on Bearings & Bearing Lubrication, and Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

does not follow, however, that there is a quantitative correlation between wear and friction.

The data reported herein were obtained in the course of experimental wear studies conducted at NACA Lewis Flight Propulsion Laboratory. The purpose of these studies was to find possible cage materials having less tendency for wear and surface failure at high sliding velocities and high temperatures than the bronze materials that are most widely used. Bronze alloys, nickel alloys, and nodular cast iron alloys were included in studies at sliding velocities to 18,000 feet per minute at room temperature and at lower sliding velocities at ambient temperatures to 1000 F. The apparatus used to obtain these data was basically a loaded, hemispherical rider of the cage material sliding on the flat surface of a rotating disk made of hardened steel. Previously reported data of this type are included in Refs. 1, 12, 13, 14, and 15.

MATERIALS. The rider materials used in this investigation and some of their properties are listed in Table I.

The thermal expansion properties of possible cage materials requires careful attention. They are especially important to high temperature bearings because, if there is appreciable difference in expansion coefficients between the cage material and race material, operating clearances may be taken-up at the cage locating surface. This is particularly true in the case of an outer-race located cage which, in an experimental bearing design, has been shown to have merit[16] for high speed operation.

The disk materials were SAE 52100 steel (hardness, Rockwell C-60 to 62) for the room temperature runs and M-10 molybdenum tool steel (hardness, Rockwell C-62 to 64 at room temperature) for the runs at elevated temperatures.

APPARATUS & PROCEDURE. Specimen preparation. The rider specimens, of the materials being investigated, were cylindrical (⅜ in. diam., ¾ in. length) and had a hemispherical tip (3/16 in. rad.) on one end. The surface of the rider specimens was finished in a lathe using light cuts and fine feeds to minimize surface cold working.

The disk specimens (either 13 in. diam. or 2½ in. diam.) were circumferentially ground using minimum material removal to produce a surface roughness of 10 to 15 rms as measured with a profilometer. These values of roughness are within the range of roughness measurements obtained on cage locating surfaces of representative rolling-contact bearings.[12]

The rider specimens were cleaned before each experiment with redistilled 95 percent ethyl alcohol. The disk specimens were carefully cleaned to remove all grease and other surface contamination. Briefly the cleaning procedure includes scrubbing with several organic solvents, scouring with moist levigated alumina, rinsing with water, washing with ethyl alcohol, and drying in an uncontaminated atmosphere of dried air or in a dessicator.

Friction apparatus. Two separate friction and wear devices were used in these experiments. Diagrammatic sketches of both the high sliding velocity apparatus using the large (13 in. diam.) disks and the high temperature apparatus using the small (2½ in. diam.) disks are presented in Fig. 1.

The high sliding velocity apparatus (Fig. 1(a)) had a disk specimen (13 in. diam.) rotated by a hydraulic motor assembly that provided accurate speed control over a range of sliding velocities of 75 to 18,000 feet per minute. The disk specimen was mounted on a flywheel assembled with its shaft supported and located by a mounting block which con-

Table I. Typical properties of materials; (data from suppliers). Key: (a) Estimate based on similar materials: (b) For comparison with SAE 52100 steel, 6.49 x 10^{-6} (77 to 300 F.); M-10 molybdenum tool steel, 6.47 x 10^{-6} (78 to 400 F.).

Material	Nominal composition, percent	Hardness, (Brinell)	Tensile strength, lb/sq in.	Elastic modulus	Coefficient of thermal expansion,[b] in./ in./°F	Source
Leaded bronze	62 Cu, 36 Zn, 1Pb, 1 Al	100-135	65-73x10^3	[a] 14x10^6	[a] 11.2x10^{-6}	Janney Cylinder Company
Iron-silicon-bronze	89-94 Cu, 2.5-4.0 Si, 1.0-2.0 Fe, 1.5-4.0 Zn, 1.0 max, Mg, 0.1 max P, 0.1 max Pb.	100-137	55x10^3	[a] 15x10^6	[a] 10x10^{-6}	Mueller Brass Company
Nodular iron	4.0 C, 2.0 Si, 1.0 Mn 0.15 P, 0.15 S, 0.016 Mg, bal. Fe.	217-253	70-90x10^3	22-24x10^6	7.4x10^{-6}	Ford Motor Company
Nodular Ni-resist	2.69 C, 2.06 Si, 1.45 Mn, 0.086 Mg, 20.34 Ni, 73 Fe	160	58x10^3	14.9x10^6	10.3x10^{-6}	International Nickel Company
Nodular Cr-Ni resist	2.98 C, 2.20 Si, 1.15 Mn, 2.36 Cr, 0.085 Mg, 20.48 Ni, 70 Fe	180	60x10^3	18.0x10^6	10.3x10^{-6}	International Nickel Company
Cast Inconel	77.75 Ni, 13.5 Cr, 6.0 Fe, 2.0 Si	175	85x10	23x10	6.4x10 (32° to 212° F)	International Nickel Company

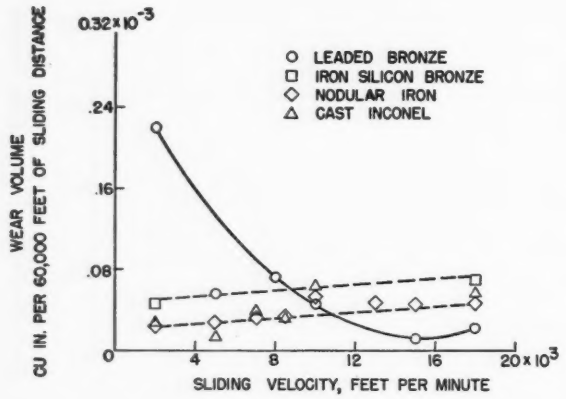

Fig. 2. Effect of sliding velocity on wear for a unit distance of sliding with several materials against hardened SAE 52100. Load, 269 grams, room temperature.

tained bearing assemblies for accurate location. A recent modification provides resistance heaters in the flywheel so that data can be obtained with combined high sliding velocities and high temperatures. Loading was accomplished by direct application of weights on the axis of the rider holder.

The high temperature apparatus (Fig. 1(b)) had the disk specimen (2½ in. diam.) mounted on a rotating shaft. The shaft was driven through a belt system by an electric motor coupled with a variable-speed power transmission device. This arrangement allowed good speed control. Loading was obtained by the use of dead weights to apply force through a pulley system. The disk and rider specimens were operated in an Inconel pot which had electric strip heaters mounted on its periphery and thus served as a furnace.

In both devices friction force was measured by restraining the motion of the rider specimen with a copper-beryllium dynamometer ring on which four strain gages were mounted. The strain gages were part of a Wheatstone bridge circuit including an observation-type self-balancing potentiometer converted for use as a friction-force indicator. Friction coefficients calculated from the friction force measurements were generally reproducible to ±0.02.

Procedure: Wear volume was calculated from the measured diameter of the wear area on the rider specimen. Wear values were checked by weight loss determinations. Incremental values of wear were calculated from rider displacement measurements during the runs or in some cases at room temperature by stopping the run and measuring the diameter of the wear area on the rider. In most cases there was oxide film or metal transfer on the disks and no attempt to measure wear of the running tracks on the disk specimens was made. The final wear-volume measurements for the rider specimens could generally be reproduced within ±10 percent in repeated experiments on a given material.

High sliding velocity wear runs were made at speeds up to 18,000 feet per minute and loads from 100 to 1200 grams. Most runs were of 1 hour duration but numerous check data were obtained for 3 hour periods. Some data were obtained with temperatures up to 650 F. at a constant sliding velocity of 8000 feet per minute. Temperatures were measured by a thermocouple installed in approximately the center of the rider at the base of the hemisphere (3/16 in. from tip of specimen).

High temperature wear runs with the small disks were all made at a sliding velocity of 120 feet per minute with a load of 1200 grams, with temperatures up to 1000 F., and had a duration of 1 hour. The temperatures were measured with a bayonet thermocouple placed adjacent to the rotating disk. The bayonet thermocouple was previously calibrated against a thermocouple installed in a rider specimen.

The influence of a lubricant on wear of the materials was determined in runs at 450 F. The lubricant was di(2-ethylhexyl)sebacate containing an oxidation inhibitor (0.5 weight percent phenothiazine). This liquid is typical of the fluids used as the base stock in some MIL-L-7808 synthetic lubricants for aircraft gas turbines. The temperature was selected because it was the maximum to which the lubricant could be subjected for a one hour period without appreciable oxidation and thermal degradation as indicated by changes of viscosity and acid number. The wear specimens were submerged in the lubricant which was at a temperature of 450 F.

RESULTS & DISCUSSION. The primary data presented are wear data although friction properties are also described. The wear data are presented as values of a wear factor for a unit distance of sliding. The criterion of wear as a function of

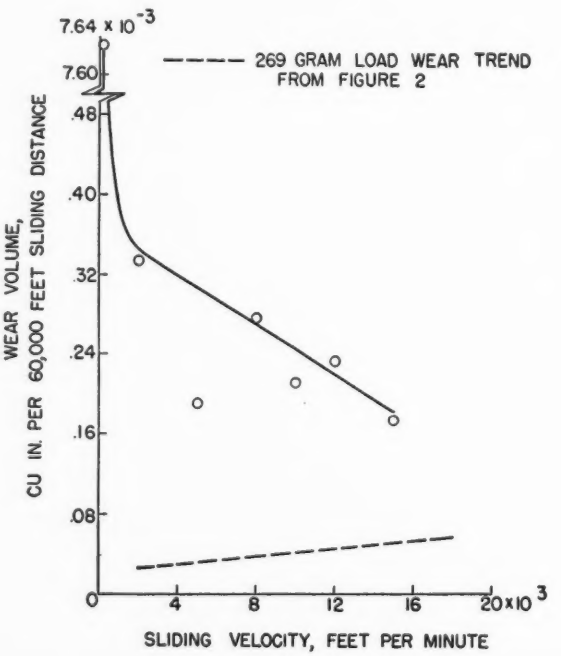

Fig. 3. Effect of sliding velocity on wear for a unit distance of sliding with cast Inconel against hardened SAE 52100. Load, 1200 grams, room temperature.

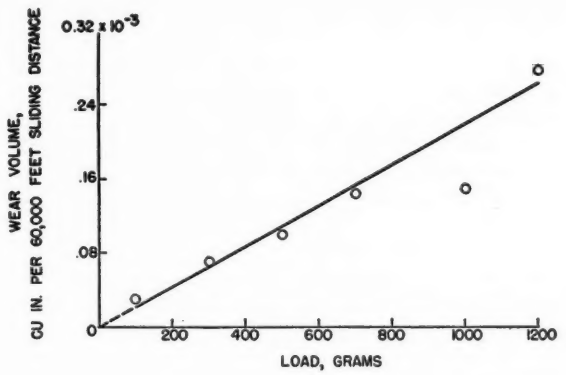

Fig. 4. Effect of load on wear for a unit distance of sliding with cast Inconel against hardened SAE 52100. Sliding velocity, 8000 feet per minute; room temperature.

sliding distance has perhaps more general support in fundamental wear studies than any other factor. The data presented herein are on the basis of sliding distance to compensate for the obvious differences between data obtained over a large range of sliding velocities (120 to 18,000 ft/min) where runs were made for unit periods of time.

Particular consideration has been given to cast Inconel in these experiments because it has essentially the same coefficient of thermal expansion as the tool steels expected to be used for the races and rolling elements of high temperature rolling contact bearings in aircraft gas turbines. Preliminary data obtained at room temperature indicates the wear of cast Inconel was essentially the same with SAE 52100 and with M-10 steel disks when the surfaces were prepared in the same manner.

Effect of Sliding Velocity. The effect of sliding velocity on wear at room temperature in air with no lubrication is shown in Fig. 2. The material showing the greatest effect is the leaded bronze, one of the commonly used bearing cage materials. The lead (1 percent) present as an alloy constituent in this material is considered to have a beneficial effect on the wear and friction of the alloy. Presumably, in this case, the effectiveness of the lead increased with the higher surface temperature that accompany increased sliding velocities. Increased plasticity of the lead would allow it to be more effective in forming a film for protecting the wearing surface. It would be expected that at some extreme condition the film would approach its melting point and wear would increase as molten film material was displaced.

The data on effect of sliding velocity on wear of iron silicon bronze, nodular iron and cast Inconel (Fig. 2) all fall within a relatively small band. Over the entire range of sliding velocities, fluctuations in wear values from the linear trend can be found; the wear trend however, shows a general increase with greater sliding velocity. For iron-silicon-bronze only two wear values are reported; these are at the maximum and at the minimum sliding velocities. The intermediate range of sliding velocities was not thoroughly studied for iron-silicon bronze but there were indications of considerable fluctuation in wear trend.

It has been suggested that wear properties of both nodular iron[12] and cast Inconel[14] derive benefit from surface films (graphitic carbon for the iron and NiO for the Inconel). It would be expected that the wear properties of these materials would vary as conditions of sliding were changed to influence the functioning or formation of the surface films. Therefore the film forming properties of nickel oxide on cast Inconel should improve with either higher sliding velocity or higher ambient temperature.

The data of Fig. 3 shows that at higher loads (1200 grams) the trend of wear volume as a function of sliding velocity for Inconel is downward or the reverse of that experienced at the lighter load (269 grams) of Fig. 2. The reduction in wear is associated with film formation properties. There was, however, visible metal transfer to the disk surface with the heavier load condition at all sliding velocities. Most metal transfer seemed to occur during the first few minutes of sliding.

The effect of varied loads in room temperature runs with Inconel at 8000 feet per minute sliding velocity is shown in Fig. 4. At these conditions, there is a linear increase in wear with load.

Effect of Temperature. The effects of ambient temperature on wear of leaded bronze, iron silicon bronze, nodular iron, and cast Inconel at low sliding

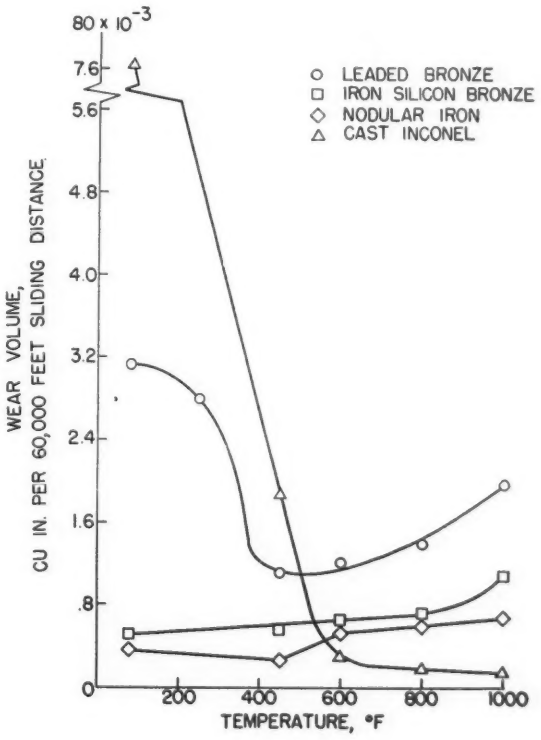

Fig. 5. Effect of temperature on wear for a unit distance of sliding with various rider materials against hardened M-10 steel. Load, 1200 grams; sliding velocity, 120 feet per minute.

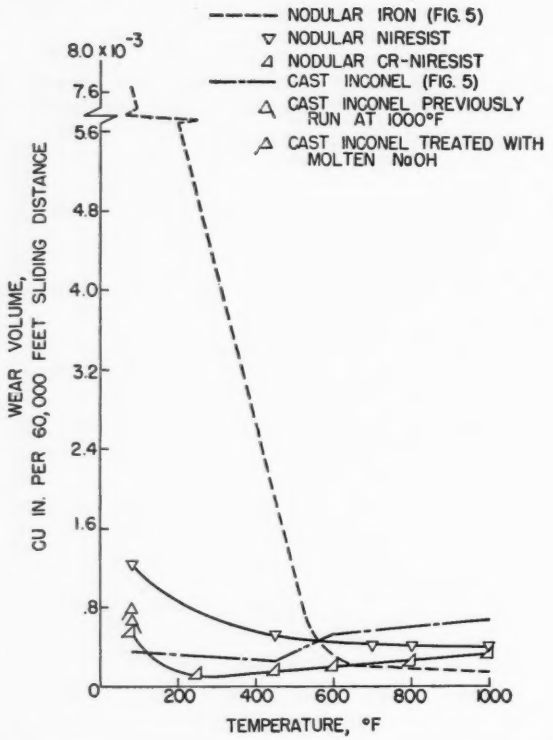

Fig. 6. Effect of temperature on wear for a unit distance of sliding with various rider materials against hardened M-10 steel. Load, 1200 grams; sliding velocity, 120 feet per minute.

velocities are shown in Fig. 5. The wear of leaded bronze decreases with increasing temperatures to 450 F. and then increases with higher temperatures. No figures are presented showing friction characteristics; with the leaded bronze there was a linear increase in friction coefficient with higher temperature levels (from about 0.3 at room temperature to more than 0.85 at 1000 F.). Iron-silicon-bronze showed a continual but slight increase in wear with temperatures up to 800 F. (Fig. 5); friction was essentially constant at temperatures below 600 F. At higher temperatures (800 to 1000 F.) wear increased more rapidly; friction coefficient changed from 0.5 at 600 F. to 0.75 at 1000 F.

At temperatures of 450 F. and lower, the nodular iron had unstable friction coefficients with values occasionally increasing from approximately 0.6 to as high as unity; this friction behavior was not, however, accompanied by high wear (Fig. 5). Friction of the nodular iron decreased slightly with increase in temperature and, at temperatures above 600 F., the friction was stable at values less than that for either of the bronzes. As shown in Fig. 5, the overall trend of effect of temperature (in the higher temperature range of 450 to 1000 F.) on wear was one of slight increase with higher temperatures.

Data at higher sliding velocities showed that cast Inconel had good wear properties under conditions where surface oxidation can occur to form protective oxide films of NiO.[14] High ambient temperatures favor the formation of oxide films. The ability of cast Inconel to provide low wear at high temperatures is shown in Fig. 5. Friction was also lowest (0.43) at the highest temperatures.

The importance of surface oxides to the functioning of cast Inconel as a slider material was demonstrated by making a run at 1000 F. in an atmosphere where oxygen availability was reduced by displacing air with argon. Wear was increased by a factor of four; only a slight increase in friction was observed. Also, as shown in Fig. 6, when cast Inconel which had previously been run for 1 hour at 1000 F. to form an oxide film was run an additional hour at room temperature, the wear at room temperature was slightly less than one tenth that obtained with a new specimen at the same condition. Further, when an unused cast Inconel specimen was heated in molten caustic (NaOH) at approximately 800 F. to preform an oxide film and was then run at room temperature the wear was also approximately one tenth that obtained under identical conditions with the untreated specimen of the same material.

The wear experienced with cast Inconel having an oxide film against M-10 at room temperature is less than one fourth of that obtained under the same conditions with the metal combination (leaded bronze and SAE 52100 steel) that has been successfully used in bearings of some full-scale engines. Newer engines use bearing cages of silver plated iron-silicon-bronze. At high temperatures, wear of cast Inconel was significantly less than that of iron-silicon-bronze. It is in the temperature range from 600 to 1000 F. that nickel alloys such as cast Inconel merit particular consideration as cage mater-

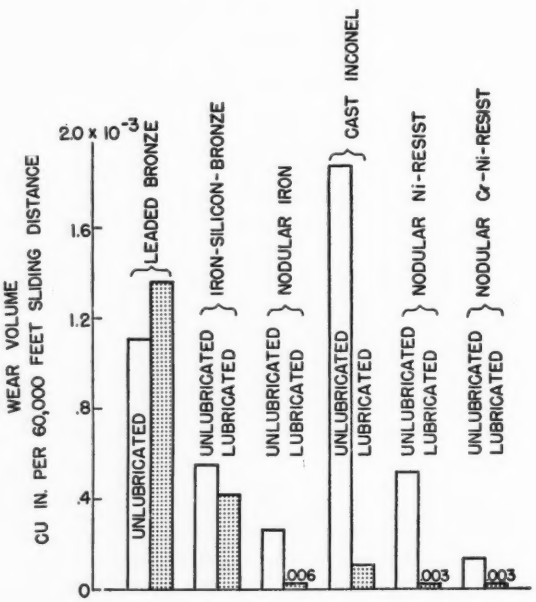

Fig. 7. Effect of lubrication with di(2-ethylhexyl)sebacate containing 0.5 weight percent phenothiazine (oxidation inhibitor) on wear for a unit sliding distance with various materals sliding on hardened M-10 tool steel. Load, 1200 grams; sliding velocity, 120 feet per minute; bulk lubricant temperature, 450 F.

ials. At lower temperatures and low sliding velocities, cast Inconel does not appear satisfactory without a preformed oxide film.

On the basis of the data on nodular iron and on cast Inconel, it seemed advantageous to obtain a high nickel content nodular cast iron. Nodular Ni-resist is such a material. It appeared to have potential merit because it was presumed that the graphitic carbon would prevent wear at low temperatures as in regular nodular iron and that a high nickel content might lead to the formation of beneficial NiO for high temperature operation. The combined effect of these surface films should give low wear at all temperatures. Data for nodular Ni-resist presented in Fig. 6 shows wear properties intermediate between nodular iron and cast Inconel at all temperatures. At room temperature, the wear for nodular Ni-resist was approximately three times that for nodular iron but at the highest temperatures it had better wear resistance than the regular nodular iron.

The producers of nodular Ni-resist suggested that the use of chromium as an alloying agent would improve the wear resistance of the material. Therefore, an alloy was obtained containing 2.36 per cent chromium (Table I) in a material that was otherwise essentially the same as the nodular Ni-resist. The addition of chromium changed the metallurgical structure from spheroidal graphite in austenite for nodular Ni-resist to spheroidal graphite and carbides in austenite for the nodular Cr-Ni-resist. As a basis for comparison, the regular nodular iron was essentially spheroidal graphite in a pearlitic structure although the total carbon in regular nodular iron was greater than for the nodular irons containing nickel.

The data of Fig. 6 show that at low sliding velocities the wear of the nodular Cr-Ni-resist was lower than nodular Ni-resist at all temperatures. Nodular Cr-Ni-resist also had lower wear than regular nodular iron at all temperatures except room temperature.

The friction of all nodular iron materials showed a decreasing trend with increasing temperatures. The nodular Cr-Ni-resist had slightly lower friction than nodular Ni-resist. At temperatures from 450 to 1000 F., the friction coefficients for nodular Cr-Ni-resist were mostly in the range between 0.64 and 0.60. At room temperature, however, the friction coefficient for nodular Cr-Ni-resist was 0.89.

Effect of Lubrication. The influence of a lubricant (di(2-ethylhexyl)sebacate plus 0.5 per cent phenothiazine) on wear of the materials was determined in runs at 450 F. Ref. 17 shows that 450 F. is above the temperature for incipient lubrication failure with di(2-ethylhexyl)sebacate lubricating steel surfaces; thus the wear data reported herein were obtained under severe conditions of boundary lubrication.

Fig. 7 shows the wear of various materials at 450 F. both unlubricated and lubricated. It is especially interesting to note that at these conditions the wear of the two bronze alloys was not significantly reduced by lubrication; in the case of the leaded bronze the lubricant resulted in increased wear. The high lubricated wear for these copper alloys may have been a result of chemical corrosion which was activated by the mechanical energy released through friction in the sliding process. It is well known that diesters cause corrosion of copper alloys at high temperatures. Also, the phenothiazine may have contributed to the corrosive wear if active sulfur compounds are formed in its decomposition. The magnitude of the bronze wear indicates that caution is necessary in the use of copper alloys for high temperature bearing parts which are to be lubricated by diester base fluids. The presence of the lubricant, however, did cause the friction coefficients for these bronzes to be about half those for the same materials when run without lubrication.

The lubricant was most effective in reducing wear of the iron base materials (Fig. 7). It was also effective in reducing wear of cast Inconel. Of additional interest, is some unreported data obtained with modified H Monel (which has been used as a cage material) showing lubricated wear about twice that for cast Inconel under the conditions of these experiments. In these lubricated experiments, ferrous base materials had friction coefficients from 0.12 to 0.15 and that of cast Inconel was 0.21.

Effect of Combined High Sliding Velocity and Elevated Temperatures. As previously mentioned, a modification of the high sliding velocity apparatus made it possible to obtain data at elevated temperatures and at high sliding velocities. Wear data for cast Inconel obtained with a sliding velocity of 8000 feet per minute at temperatures up to 650 F. are shown in Fig. 8. By comparison with the data of Fig. 5, it can be seen that the trend of wear to decrease with of increased temperatures prevails at

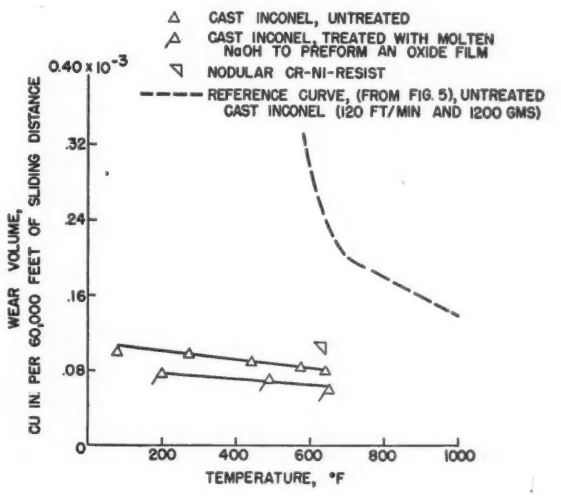

Fig. 8. Effect of temperature on wear for a unit distance of sliding with cast Inconel and nodular Cr-Ni-resist against hardened M-10 steel. Sliding velocity, 8000 feet per minute; Load, 500 grams. Note: 80 F. data for untreated cast Inconel obtained with hardened SAE 52100 steel.

both low (Fig. 5) and high (Fig. 8) sliding velocities.

Data were also obtained with cast Inconel pretreated to have a NiO film. The results show (Fig. 8) that even under conditions favorable to the natural formation of NiO the untreated and treated cast Inconel show comparable wear. With the chemically pretreated specimen, there was no visible evidence of metal transfer to the steel disk. Using untreated specimens slight metal transfer occurred at most conditions during the initial part of the wear runs. In a preliminary check, it was found that a preformed oxide film obtained by heating cast Inconel in air for two hours at 1200 F. was not as effective in reducing metal transfer as the film obtained by chemical treatment.

Data obtained with nodular Cr-Ni-resist (Fig. 8) at the combined condition of high sliding velocity and elevated temperature shows wear comparable with cast Inconel. The amount of wear for leaded bronze was much greater than for cast Inconel.

CONCLUDING REMARKS. Both sliding velocity and temperature had considerable effect on wear of unlubricated surfaces. The influence of chemical effects was accentuated by the extreme range of physical variables such as temperatures. The ability of oxide films to reduce wear was demonstrated most effectively for cast Inconel sliding unlubricated against hardened steel. Either naturally formed oxide films developed during sliding at high temperatures and also at high sliding velocity or oxide films formed by chemical pretreatment prevented visible metal transfer and markedly reduced the volume of wear of the cast Inconel. The corrosive influence of the diester lubricant on copper was considered responsible for the high wear of lubricated bronze alloys at 450 F. One bronze alloy had more wear at 450 F. when lubricated than when run without lubrication.

Cast Inconel pretreated to have a black oxide film is believed to have promise as a possible cage material for high temperature rolling contact bearings. It has good wear and surface failure resistance that improve with severity of operation and has essentially the same thermal expansion coefficient as a tool steel (M-10) used for high temperature bearings. Because the oxide film is reformed during operation, this type of film does not have the disadvantage of limited life that is frequently associated with the use of electroplated films such as silver plating. The oxide film, however, does not give low friction.

Nodular iron and nodular Ni-resist materials had less wear at elevated temperatures than bronze alloys. The wear of nodular Ni-resist (20 per cent nickel) was less at temperatures above 600 F. than that of regular nodular iron. A nodular Ni-resist containing chromium (2.36 per cent) had less wear under most conditions than the other nodular irons. Friction values for the nodular Cr-Ni-resist and the nodular Ni-resist were generally higher than that for the regular nodular iron. Wear of nodular Cr-Ni-resist is comparable with that of cast Inconel or cast Inconel with a preformed oxide film under most conditions and has better response to lubrication by diesters. Nodular-Cr-Ni-resist, however, has much greater thermal expansion coefficient than M-10 tool steel.

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★ **2nd Annual ASLE-ASME Lubrication Conference, Oct. 10-12** ★

Lubrication Problems In The Steel Industry (Panel)

T. A. Marshall, Panel Chrmn.*

At the ASLE 1954 Annual Meeting in Cincinnati a panel meeting was held to discuss lubrication application problems in the steel industry, and their solution. The panel included two members of the steel industry, **C. A. Bailey** of U. S. Steel Corp., and **C. T. Lewis** of Republic Steel Co.; two representatives from equipment manufacturers, **S. S. Rickley** of Morgan Construction Co., and **B. J. Lamport** of United Engineering & Foundry Co.; and two steel lubrication specialists from petroleum companies, **G. H. Davis** of Shell Oil Co., and **M. S. Clark** of The Texas Co. A lubrication engineer from a British steel concern, **G. D. Jordan** of Samuel Fox & Co., Ltd., was a guest member. After brief introductory remarks by the panel members the session was devoted to answering questions from the audience. The questions and answers dealing with the more important problems are herewith presented in condensed form.

Q. Are inhibited turbine, transformer, and hydraulic oils economically worthwhile?

A. Yes, except in cases where excessive contamination occurs. (Clark)

A. In one case when a 1000 Kva transformer was located near a blast furnace and was operated at about 170 F., uninhibited oil lasted about 24 months, while inhibited oil lasted about 84 months. (Jordan)

Q. When inhibited hydraulic oils are used, are copper and copper alloys as satisfactory as steel for tubing and tubing fittings?

A. No, because possible catalytic action may shorten inhibitor life. (Clark)

Q. Is there a need for a fire-resistant hydraulic oil in steel mills?

A. Yes, where possible danger exists. However, poor lubricity of such materials must be considered under high pressure applications. (Lewis)

Q. What physical or chemical tests can be used to determine the rate of additive depletion of detergent oils?

A. Only a limited number of tests are available. Oil companies run full-scale engine tests which furnish significant results but which are expensive. The Bureau of Mines has developed a hot strip test in which the strip is maintained at 450 F. with oil flowing over it. While detergency remains, the oil flows leaving the strip bright. When the detergency is depleted, the oil sticks and burns. This test has not seen much service. (Bailey)

Q. What is a good test for water separation in circulating systems?

A. The demulsibility test is significant when used on new oils. Once the oil has been placed into service, results from this test are meaningless. The water content of the oil can be checked to determine the effectiveness of water separation. (Clark)

Q. In view of the varying speeds and loads under which a cold reduction mill operates, why does the manufacturer specify such a narrow viscosity range for the lubricating oil used in these bearings?

A. This question can probably best be answered by picking out an example mill. Assume a 4-High Reversing Cold Mill with a speed range of 600-1200 FPM. For that mill we would probably specify an oil with a viscosity of 2400-2500 seconds Saybolt Universal at 100 F. Now that is a narrow spread, just 100 seconds. But with this oil supplied to the bearings at 100 F., we know the mill will operate satisfactorily at the 1200 FPM top speed without excessive heat generation. We will also have an oil with a heavy enough viscosity to establish a complete oil film with complete metal to metal separation, even when the mill is operating below base speed.

If the specified viscosity range was extended and we assume a 10 F. error in supply temperature, it is easy to see that adequate lubrication for all speed conditions is very doubtful.

Q. Has Morgan Construction Co. considered recommending bearing oils of various viscosities so that the customer can choose a viscosity for whatever temperature he wants to operate?

A. We are primarily interested in the viscosity of oil that goes to the bearing and not in particular the temperature at which it is supplied within reasonable limits. We know that bearing metals have a critical operating temperature. Therefore, we limit the supply temperature so that the resulting operating temperature will be well below the critical temperature of the bearing metal. 100 F. is assumed

*Steel Company of Canada, Ltd., Wilcox, Hamilton, Ontario, Canada.

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in most cases because it meets this condition; it is a temperature that can be readily obtained whether it is winter or summer operation. If, for example, we specified 120 degrees it might be tough to get the oil up to 120 degrees in the wintertime and, conversely, if we specified 80 it would be tough to hold it to 80 degrees in the summer heat. In special cases the oil viscosity is adjusted within reasonable limits to allow for unusual heating or cooling conditions.

Q. Morgan Construction Co. recommends straight mineral oil for Morgoil bearings. Some users have experienced scoring due to lack of proper film strength in starting operation. Will you comment on the utility of non-corrosive EP lubricants to eliminate this potential difficulty?

A. No EP requirement exists because the bearing operates under full hydrodynamic lubrication. As such, straight line contact exists in the bearing prior to shaft rotation. As soon as rotation is started an oil film is established in much less than one revolution of the journal. The scoring more likely was caused by dirt or other contamination. (Rickley)

Q. What is the best procedure to prevent corrosion between a Morgoil bearing journal and tapered roll neck?

A. Corrosion should not occur because a continuous supply of bearing lubricant goes through the journal to the neck of the roll. If a rust-like material occurs, it may have been caused by high water content of the oil. Another possible condition might consist of black spots on the neck. They are quite common and are so-called "pressure oxide" caused by the extremely high pressures of steel on steel. These spots are not burned oil and are in no way detrimental to bearing or roll neck operation. (Rickley)

Q. What is the maximum water content that may be in Morgoil Bearing oil for safe operation?

A. No percentage can be stated. As long as a homogeneous mixture exists, lubrication will be maintained. However, water in oil indicates high oil consumption and degrades oil performance. (Bailey)

A. In one hot strip mill the bearing oil was found to have 35 per cent water in it and was still operating satisfactorily. This is not recommended practice. (Rickley)

Q. Would it not be advisable for the designers and builders of circulating oil systems to design twin compartment reservoirs for capacities of 1000 gallons or higher?

A. Twin tanks are advisable whenever water can get into the EP oil used in drive or table systems. A single tank with a dividing plate is not recommended because of heat transfer. (Lamport)

Q. If spur, helical, or herringbone gears were made properly — both dimensionally and metallurgically — to transmit the desired load, would there be any need for EP lubricants for such drives?

A. Not as such. This is proven by numerous examples where EP lubricants cannot be used, such as drive units which include friction clutches or turbine applications in which the oil lubricates both the drive bearings and the reduction gears. (Lewis)

A. In some cases, although the gears were made properly, the operator increases the speed and load considerably over the rated capacity. In such cases an EP oil is required for satisfactory gear performance. (Clark)

Q. What are the desirable and undesirable characteristics for lithium greases?

A. The characteristics of lithium greases include very good mechanical stability, high water tolerance, high melting points, ease of pumping, and multipurpose uses. As disadvantages, separation in some central systems and some rusting have been reported. Steps are being taken to correct these factors. (Davis)

Q. Has the introduction of lithium EP lubricants for mill rolling contact bearings contributed to the life and load-carrying capacity as compared to the conventional lime leaded EP lubricants?

A. Lithium grease is more satisfactory at high temperatures. We have found over 50 per cent water mixed with it and the lithium grease still lubricated satisfactorily. (Davis)

A. In one mill on the Continent only lithium EP grease operated satisfactorily. In a 17-stand continuous roughing mill with spherical roller bearings, lithium EP lubricants are used with excellent results. (Jordan)

Q. What is being done by oil companies to determine the type of grease least likely to separate in centralized systems? What is being done to improve the situation?

A. Considerable work is being done on lithium and inorganic gel stabilized greases as well as the more conventional types. Some of these new products are already on the market and still better ones are on the way. (Bailey)

A. Although grease separation can be a serious problem, a small amount of separation is desirable for lubrication at start-up. (Clark)

A. Oftentimes the dispensing equipment has considerable effect on grease separation. These factors, along with the effects of base oil viscosity and additives, are all being investigated. (Davis)

Q. Is the use of grease considered necessary in the operation of fabric bearings? Have there been cases where the use of grease or other lubricant has proved detrimental in fabric bearings?

A. Two 45-inch slabbing mills with fabric bearings have been installed by us within the last year. The mills were equipped with automatic grease feeders which supplied grease to each bearing every time the mill was shot. The feed to the roll neck bearings on both mills were disconnected and only a small amount of grease was supplied during shutdown to prevent corrosion. One mill operated satisfactorily under these conditions, while bearings in the other mill failed frequently until the continuous grease feed was resumed. We feel it to be very

important that a coating of grease be maintained on the roll neck during any shutdown to prevent corrosion. By feeding the grease continuously, a protective coating is assured even during a breakdown. Considering the protection, the cost of the grease is minor. (Lamport)

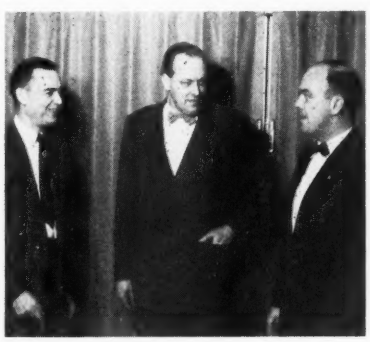
A. In the United Kingdom a lubricant has been developed for fabric bearings which is composed mostly of lanolin and a sodium potassium base. It is a paste and is completely water soluble. It is fed into the water supply, not directly to the mill. Considerabe improvement over plain water was noted. (Davis, Jordan)

Q. What is the proper frequency for lubricating coke quench-car axle bearings of the rolling contact type? What type of grease should be used?

A. This is primarily a seal problem as the seals must keep out the corrosive chemicals in the quenching fluid. Efficient seals should function satisfactorily for six months. With efficient seals a high melting calcium or lithium grease should give one year's service. (Lewis)

Section News

Notice to ASLE Section Secretaries: To insure early and adequate coverage of your Section's activities in "Lubrication Engineering," all Section News copy (including Summaries of papers presented, Election of Officers, Photographs, etc.) should be received by the ASLE National Office 1½ months prior to the month of publication. (NOTE: Photographs submitted for possible publication should not include more than five persons; each person must be identified.)



PITTSBURGH, February Meeting. Annual National President's Night, with J. Boyd (ASLE Past-President), Westinghouse Research Laboratories, presenting a paper entitled "The Principles of Lubrication in Journal Bearings & Pivoted-Pad Thrust Bearings." (Pictured left to right, Section Chrmn. C. E. Trautman, Mr. Boyd, & W. P. Youngclaus, Jr., ASLE Administrative Secretary.)

March. A. L. Hartley, Chief Metallurgist of the Cincinnati Milling Machine Co., spoke on the subject "Advantages of a Carefully Planned Lubrication Program." (Submitted by D. W. Howard, Publicity Chrmn.)

BALTIMORE, April. Dr. H. E. Mahncke, Westinghouse Research Laboratories, presented a paper entitled "Viscosity Classification Systems," in which the need for and the usefulness of such a system was discussed, as well as various units used to designate viscosities. (Submitted by J. E. Buchanan, Sec'y.)

CHICAGO, March. H. W. Winkler, Vice-President and Director of Development & Research for The Brooks Oil Co., presented a paper entitled "Influence of Temperature on Petroleum Lubricating Grease."

April. Dr. H. J. Dawe, Director of Research for Acheson Colloids Co., presented a paper entitled "Characteristics & Performance of Solid Lubricants," in which he pointed out that lubrication is tied in closely with the theories of metallic friction. 'The welding-shearing theory is the most generally accepted one to explain the greater part of metallic friction and is of considerable importance in considering machine lubrication. Even in systems designed for hydrodynamic lubrication, any lack of adequate fluid lubricant or too heavy a load results in boundary conditions. Mixed types of lubricants—liquids plus solids — help solve the problems of boundary lubrication. The properties of graphite or molybdenum disulfide make these materials especially advantageous for most conditions where lubrication is difficult. In colloidal form these solid lubricants are dispersed in fluid lubricants or volatile carriers and are carried to all working parts of machinery, exposed or unexposed, in regulated amounts.' (Submitted by A. B. Wilder, Sec'y-Treas.)

CONNECTICUT, October '54. E. W. Fisher, The Garlock Packing Co., presented a paper entitled "Oil Seals & Mechanical Packings."

November '54. C. D. Fleming, Socony-Vacuum Oil Co., spoke on the subject "Functions of Cutting Fluids in Machining Metals."

January. C. R. Gillette, New Departure Div. GMC, presented a paper entitled "Rust Prevention in Manufacturing Operations."

February. W. S. Wise, Director of the Connecticut State Water Commission, spoke on the subject "Stream Pollution."

March. Election of Officers (see ASLE Directory), followed by an address by F. R. Archibald, Arthur D. Little, Inc., entitled "Viscosity Flow Problems of Lubrication." (Submitted by L. M. Edwards, Publicity Chrmn.)

KINGSPORT, March. J. Boyd (ASLE Past President), Westinghouse Research Laboratories, presented a paper entitled "Sliding Type Bearings," in which the action of journal bearings and pivoted pad thrust bearings was demonstrated. (Submitted by J. E. Fleenor, Chrmn.)

LOS ANGELES, April. A. I. Sippola, Socony-Vacuum Oil Co.,

(Continued on p. 199)

Sliding Friction Test For Metalworking Lubricants

by W. J. Wojtowicz*

Lubricants used in drawing, stamping and forming may be evaluated by a simple sliding friction test. Measurements are made at slow speeds, 4 inches per minute, and moderate pressures, 5000 to 10,000 psi. Flow of metal is correlated with the average kinetic coefficient of friction; welding with a progressive increase of the friction force. Significant differences are observed when lubricants are tested on different metals. Aluminum, titanium, enamel stock, stainless, galvanized and cold roll steel have been investigated with lubricants such as: petroleum oils, chlorinated paraffins, waxes, dry soap films, and proprietary emulsions. The observed trends are consistent with conclusions derived from production tests as well as previous investigations. Anomalous behavior of lubricants, particularly dry films and chemical coatings is discussed on basis of friction data.

The choice of lubricants as related to coldworking of metal has been a source of considerable concern. The selection of adequate lubricants has been a challenging problem because of the number of factors which need to be considered before ascertaining the desirability of a specific product. Foremost of these problems has been the ability of a given product to allow the metal to flow properly and to minimize galling and scoring of the tools and/or the fabricated parts. The former quality is associated with reduction or control of friction, the latter with inhibition of metallic welding.

Laboratory methods for evaluating the quality of drawing lubricants have been suggested by many investigators.[1,2,3,4,5,6,7,8] Numerous types of tests are in actual daily usage and these assist materially in standardization and quality control of metalworking lubricants. A majority of the methods fall into one of two categories: (1) measurement of energy consumed in performing a coldworking operation, and (2) direct friction measurements. Each of these techniques has its applicability, although the latter would seem more desirable because it leads to an understanding of the basic problems encountered in this specialized field of lubrication.

Metalworking can be divided into three classes representing operations such as: (1) deepdrawing at pressures approaching 10,000 psi, (2) ironing of metal at 100,000 psi and (3) most severe cold extrusions, which may reach unit pressures in excess of 300,000 psi. Within each of these categories can be found a wide range of speed, varying from several inches to as much as a mile per minute. The possibility that a single test method could be applicable for all types of problems is very unlikely, especially in view of the behavior of lubricants at high pressures[9,10] and/or high surface speeds.[11] The following study is accordingly limited to problems arising in deep drawing of metals at pressures not in excess of 10,000 psi and at relatively slow speeds.

TEST PROCEDURE. A modification of the Heinz Friction Test method[4] was adopted for this investigation. Fig. 1 shows a schematic drawing of the technique employed. A lubricated test strip was placed between a pair of flat polished dies which were loaded by means of a calibrated torque wrench acting on a screw. The entire assembly was rigidly mounted on a tensile machine, which supplied the necessary motivating force. Normally tests were made with SAE 1010 CR strips which had been degreased with benzene. Special investigations were necessarily made with test strips typical of the metal which was being coldworked. The lubricant was applied to the test strip by swabbing, rolling, brushing or dipping in accordance with conven-

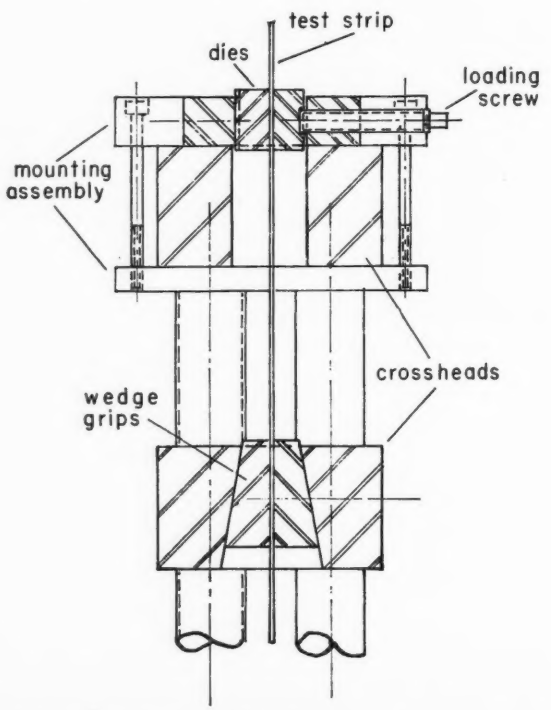


Fig. 1. Schematic illustration of test assembly.

*H. A. Montgomery Co., Inc., Detroit, Mich.

(This paper was sponsored by the ASLE Technical Committee on Fluids for Metal Working, and presented at the ASLE 10th Annual Meeting, Chicago, April 15, 1955.)

tional practice. 2" x 2" flat, hardened, tool steel dies were used. These were hardened to Rockwell C-62, ground and lapped. To insure reproducible results the dies had to be broken-in. Generally, this was accomplished by carrying out approximately 50 friction tests under a 20,000 pound load. A carbide scraper was used to remove gross pickup and frequent lapping of the dies was necessary to maintain a desirable high quality surface.

Calibration of the torque wrench was carried out with the aid of strain gauges. (An alternate method consisted of suspending the fixture assembly between the base and the moving head of the Riehle Tensile Tester, Model PA 60, and determining the compressive load induced by the loaded screw.) The applied load was in the range from 4,000 to 20,000 pounds, corresponding to 1,000 and 5,000 psi pressure respectively on a test strip 2 inches in width. A unit pressure 10,000 psi could be obtained by using a one inch test strip. Reproducibility of loadings was ± 5%. Oscillographic scanning during test runs indicated only a negligible variation in the applied load. Speed of the test draw was four inches per minute.

EVALUATION OF TEST DATA. Preliminary friction studies at unit loads as low as 4,000 pounds indicated that the friction force during a test run was not always constant. The observed pattern was similar to that noted in work previously described.[7] Poor lubricants invariably led to a rapid rise of the friction force. In order to properly evaluate lubricants it was necessary to determine the average friction force. The latter value was obtained from an autographic recording which corresponded to a test run of four inches. A series of three consecutive tests was sufficient to establish the magnitude of the coefficient of friction and the scoring factor.

Recordings with positive slope are invariably associated with lubricants which have poor anti-scoring qualities. Negative slopes, on the other hand, are typical of lubricants which are known to possess excellent anti-scoring properties. Among the latter group would be found wax films, dry soap films and chemical coatings.

The deviation of the coefficient of friction in consecutive tests was within 5% of the mean value. Where severe scoring was encountered, the average deviation was as large as 10%. Even under the latter conditions, the values which were obtained were significant, because the spread in the coefficients of friction between a good and a poor lubricant could vary by a factor of 50. Depending on the type of lubricant, the metal, and the loading, the coefficient of friction was as low as 0.004 or greater than 0.250. The upper value did not represent a limit because, under the prevailing test conditions, the friction force corresponding to the latter coefficient was in excess of the tensile strength of the normal test piece.

The scoring factor was not nearly as well defined. Its magnitude was designated as a ratio of the average friction force to minimum observed friction force. Deviations of 20% from mean were frequently noted. The absolute value could therefore be used only as a qualitative measure of the anti-scoring properties of the lubricant. The range of observed values was from —2 to +4, the algebraic sign denoting the slope of the friction force curve.

APPLICABILITY OF TEST METHOD. The friction test method has correlated data obtained with a previously described quality control test.[7] The simplicity and ease of interpretation of the data in this test, however, made it far more desirable. It has been found useful in evaluation of many pertinent variables encountered in drawing, stamping and forming operation. Among the problems studied were: (1) significant physical properties of metalworking lubricants, (2) specificity of lubricants for different classes of metals, (3) proper usage of lubricants. Serious complications arose when this test was contemplated for evalua-

Table I. Coefficient of friction data; load, 20,000 lbs. Key: (a) Load, 5,000 lbs.; (b) Severe chattering, galling, test discontinued.

Metals Key: (A) SAE 1010 CR; (B) Stainless steel; (C) Galvanized iron; (D) Enamel iron; (E) Aluminum; (F) Tern plate; (G) Titanium.

Lubricants Key: (1) Petroleum oil, 5000 SSU @ 100 F.; (2) Chlorinated paraffin, 50% Cl.; (3) Heavy duty pigmented emulsion; (4) Heavy duty non-pigmented emulsion; (5) Beeswax; (6) Dry soap type film.

Metal	Lubricant					
	1	2	3	4	5	6
A	0.025	0.012	0.021	0.018	0.086	0.143
B	0.019	0.008	0.101	0.095	0.089	0.101
C	0.160	0.066	0.180	0.200	0.124	0.116
D	0.130	0.043	0.210	0.220	0.110	0.134
E	0.005	0.004	0.023	0.005	0.120	0.120
F	0.103	0.057	0.060	0.060	0.081	0.108
G	0.240[a]	0.012[a]	0.160[a]	(b)	0.090	0.140

Table II. Scoring factor (Σ) = Friction Force (Average)/Friction Force (Minimum); load, 20,000 lbs. Key: (a) Load, 5,000 lbs.; (b) Severe chattering, galling, test discontinued; (c) Negative slope. (See also, Metals & Lubricants Keys, Table I.)

Metal	Lubricant					
	1	2	3	4	5	6
A	1.0	1.0	1.3	2.4	1.0	1.0
B	-1.1[c]	1.4	1.0	4.0	1.0	1.4
C	-1.1[c]	1.3	2.0	1.1	-1.1[c]	1.3
D	1.0	2.2	1.0	1.1	-1.1[c]	1.1
E	1.5	2.9	1.5	3.5	1.2	1.1
F	1.0	1.1	2.0	3.5	1.2	1.1
G	3.5[c]	1.2[a]	4.0[a]	(b)	-1.1[c]	1.1

tion of lubricants for wire or tube drawing, or cold extrusion. The first of these problems was the lack of a suitable metal test specimen, the second was the necessity to interpret the behavior of lubricants at very high pressures and speeds from data obtained in the laboratory under relatively mild conditions. Preliminary attempts to extend the same technique to pressures of 200,000 psi, as outlined in the Heinz report,[4] were not satisfactory.

TEST DATA. One of the most important physical properties of drawing compounds is its viscosity.[12,13] In many cases it is the only factor which needs to be considered. Fig. 2 shows the dependency of friction on viscosity. Petroleum oil blends were prepared to cover a range of viscosity from 63 to 4400 centistokes at 72 F., corresponding approximately to Saybolt Seconds Universal of 100 to 3500 at 100 F. Tests were made with standard two inch test strips under 10,000 pounds load. Under these conditions there was but negligible scoring. Initially, friction decreases rapidly with an increase in viscosity. It subsequently becomes constant and independent of viscosity. The critical viscosity at which this transition occurs is dependent on the type of metal used, the speed and the loading. Under the conditions outlined, derivatives of polyalkylene glycols behaved identically to petroleum oils.

One of the most troublesome problems encountered in selection of lubricants is their occasional anomalous behavior. The difficulty can be directly associated with the nature of the metal being fabricated. Tables I and II present data collected in a series of tests made with seven different metals and six types of lubricants. It was evident that not all classes of lubricants were adaptable for cold working some of these metals. In a few cases, the choice was narrowed down to only one type of a product. The selection was determined by the function which the lubricant could adequately fulfill. This could have been either reduction of friction or prevention of scoring. While there may be an intangible relation between these two properties, it could not be assumed that a given product would fulfill

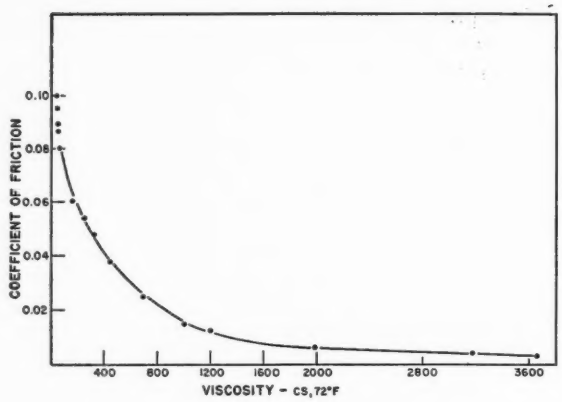


Fig. 2. Effect of viscosity on friction force.

both of these functions equally well. Viscous oil type compounds were most successful in reduction of friction, whereas, dry soap or wax films were preferred for elimination of galling and scoring.

The applicability of the six types of lubricants is summarized in Table III. The conclusions arrived at from simple friction tests are in agreement with the general practice which is exercised in selection of lubricants for coldworking of the indicated metals. What appears as an anomalous behavior on the part of the lubricants is, in effect, a phenomenon associated with the friction occurring during deep drawing of metal.

The behavior of titanium in these tests was in accord with results obtained by other investigators.[14] Severe galling of dies was experienced with viscous oils or emulsified lubricants even at moderately low loads of 5,000 pounds. To reduce friction effectively under these mild conditions required the usage of viscous chlorinated oils. To overcome galling, dry soap or wax films were necessary. Equally successful would be any type of a bonded solid interface over which would be applied a viscous oil-type lubricant.

The frictional properties of a given class of metal can vary appreciably. This variation has, at least in one case, been accountable for trouble experienced in fabrication of welded tubing. The metal in question was silicon killed 1010 cold rolled steel. Table IV lists the coefficients of friction for specimens obtained from four different suppliers.

Table III. Selection of lubricant. Key: (X) Reduces friction effectively; (XX) Decreases galling and scoring; (XXX) Reduces friction and decreases galling and scoring. (See also, Metals & Lubricants Keys, Table I.)

Metal	Lubricant					
	1	2	3	4	5	6
A	XXX	XXX	XXX	X	XX	XX
B	XXX	XXX	X	-	XX	XX
C	XX	XXX	-	XX	XXX	XXX
D	XX	X	XX	XX	XXX	XXX
E	XXX	X	XX	X	XX	XX
F	XX	XXX	X	X	XX	XX
G	-	XXX	-	-	XX	XX

Table IV. Metal: Silicon killed 1010 CR steel; Lubricant: Viscous oil, 40,000 SSU @ 100 F.; Load: 20,000 lbs.

Supplier	f
A	0.020
B	0.023
C	0.031
D	0.037

Table V. Metal: SAE 1010 CR steel; Lubricant: 70% borax, 30% high titer soap; Load: 20,000 lbs.

	f	Σ
Original film	0.140	1.1
Humidified film	0.065	3.0

Welded tubing made with type D metal was difficult to cold work. The problem was solved by reformulating the lubricant to accommodate the poorest quality metal.

Dried soap films have been used extensively in cold working of metal. Considerable variation has been observed in their performance. This trouble has been generally associated with the hygroscopic character of the dried film. Table V presents the pertinent frictional properties of a dried soap/borax film before and after exposure to an atmosphere of 90% relative humidity at 80 F. for 24 hours.

The dry film possessed good anti-scoring qualities, whereas, the humidified film had better anti-friction qualities. The plasticizing effect of the absorbed moisture in the latter case seriously affected the anti-scoring properties of the film. The observed differences in the physical properties could account for some of the problems encountered in the use of soap films.

CONCLUSIONS. Laboratory evaluation of deep drawing compounds may be carried out with a simple sliding friction test. The outlined procedure gives pertinent information regarding the ability of a lubricant to decrease friction and to inhibit metallic welding during deep drawing. Experimental data confirms practical experience and indicates that there exists a specificity between lubricants and metals.

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ACKNOWLEDGEMENT. Titanium metal used in these investigations was contributed by the Mallory-Sharon Titanium Corp.

Current Literature

"'dag' Colloidal Graphite for Electronic & Electrical Applications," Bulletin 433, describes how 'dag' colloidal graphite (a high-purity electric-furnace graphite processed to colloidal size and dispersed in a wide variety of fluid carriers), is used for coating the insides of cathode-ray tubes in order to prevent stray electrons from reflecting back into the electron beam. Also mentioned is the utilization of graphite films for increasing thermal radiation and for decreasing secondary emission, "back" omission, and undesirable photo-electric effects in vacuum tubes. Methods for improving the efficiency of copper oxide rectifiers, improving connections between graphite coatings and ground wires, and attaching filaments to lead-in wires of various types of carbon filament lamps by means of collodial graphite dispersions, are presented. Procedures are given for electro-static shielding of vacuum tubes, television cabinets, hearing-aid cases, and tape recorder cases. Static elimination in drive belts, dust collector bags, instrument windows, and airplane surfaces is described, and lubrication of electrical devices which must operate for long periods of unattended service is explained. (Acheson Colloids Co., LE-11/3, 1630 Washington Ave., Port Huron, Mich.)

Lubrication Chart. A handy wall chart containing lubrication recommendations for many industrial needs is now available, covering such applications as hydraulic systems, spindles, air compressors, reduction gears, electric motors, oven conveyors, and other general uses. Printed on tough varnished stock for protection from smudging or handling, this chart can be displayed in prominent places in the shop to help eliminate lubricant misapplications. (E. F. Houghton & Co., LE-11/3, 303 Lehigh Ave., Philadelphia 33, Pa.)

"Recommended Practices for Lubricating Automotive Front Wheel Bearings," a step-by-step procedure developed with the cooperation of more than 250 experts in the automotive and oil industries, has been released that includes 28 photographs on how to disassemble, clean, relubricate, and adjust passenger car and

(Continued on p. 205)

Maintenance Of Water-Soluble Machining Coolants During Use

by J. A. Prevel*

The use of soluble machining coolants has increased tremendously in recent years due largely to the introduction of so-called heavy-duty soluble coolants. Also many machines have been designed to use soluble machining coolants which was not possible before. The use of soluble machining coolants has many advantages in machining operations, but care must be taken to control the coolant to insure best results. Accurate, but practical methods of control in the shop can extend the life of the coolants in the machines. Methods of control include both physical and chemical, depending on the conditions encountered.

The use of water-soluble machining coolants has increased tremendously in recent years. The introduction of heavy-duty water-soluble coolants has been one of the main reasons; also many machines are now designed so that water-soluble machining coolants can be used in them.

The main functions of cutting fluids are cooling of tool and workpiece, and lubrication between tool and chip; the latter prevents welding of the chip to the tool. In addition, a water-soluble oil should protect the work and tool against rust, have a mild odor, be non-toxic, and be stable over a reasonable length of time when prepared into emulsions.

Soluble oils are designed so that the oil can be mixed with water at the user's plant to form the emulsion. Since this is the case, the soluble oil should be a properly selected and blended material which will emulsify with waters of varying hardness.

In addition to emulsifiers, the compounder incorporates materials to the base oil which possess bactericidal properties, anti-corrosion properties, and extreme-pressure properties in the case of heavy-duty soluble oils.

Proper mixing of the soluble oil and water is necessary if a good emulsion is to be formed. The oil should always be added to the water and mixed well. If the reverse is done, an inverted emulsion may be formed which is quite unsatisfactory. In cases where very hard water is available it may be advisable to soften the water by some convenient method, unless the soluble oil being used was designed specifically for use in hard water. The difficulty of forming good emulsions with hard water is due to the amount and nature of the salts dissolved in the water which form insoluble soaps when these salts react with the emulsifying materials. The effectiveness of the soluble oil emulsion is greatly impaired.

The application of soluble oils in a machine involves a large number of variables, such as type of water used, plant cleanliness, preparation of emulsion, condition in machine, type of metal, etc. The following corrective measures are helpful in eliminating certain adverse conditions:

(1) Remove the oil blanket on the surface of the emulsion to prevent insulation from the air.

(2) Remove chips or fines from machines at frequent intervals. The use of filters of all types are very useful to accomplish this and most generally pay their own way in money saved on labor to clean machines and longer coolant life in machines.

(3) Clean the machine coolant reservoir and system periodically with a good machine cleaner.

(4) Aerate the emulsion frequently to remove dissolved gases which collect in machines that are not used continuously.

(5) Maintain check on pH of the emulsion. Most emulsions have a pH of 8.0 and 9.0 indicating that they are basic. A drop in pH below 7.0 usually indicates that the emulsion is going acidic in the machine and will eventually separate, as most are not stable when they become acidic.

(6) Check emulsion concentration at frequent intervals. Generally there is a definite concentration at which the job works best, and changes in concentration usually reflect in shorter tool life. The method generally used is to take a graduated cylinder or soluble oil flask and fill to desired level. Add a few drops of sulfuric or hydrochloric acid and agitate well. The oil will rise to the surface, and the concentration can be calculated or read on scale depending on whether a cylinder or flask was used.

Rancidity and offensive odors sometimes develop during the use of soluble oils. Generally these conditions exist when the bacteria count is high. These bacteria may come from sweepings off the floor, some well waters which contain bacteria which thrive on absence of oxygen, from workers who wash hands in coolant, and many other ways. If

*General Motors Corp., Buick Motor Div., Willow Springs, Ill.

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these bacteria have enough time to go to work a rancid emulsion develops.

During weekend shutdowns, and especially if the emulsion is covered with an oil blanket, there is greater chance for the development of offensive odors. In most cases, hydrogen sulfide gas (rotten egg odor) is evolved. When emulsions go rancid, or develop offensive odors, the best thing is to drain machine of the coolant, add strong machine cleaner with a germicide, and clean machine thoroughly. Investigations should also be started to find the cause of the rancidity.

The following system, which has been found to be satisfactory, was developed through trial and error. At first the responsibility was on the foreman to request draining and cleaning of machines. However, it was apparent that a better system had to be devised for this particular plant. The foreman of coolant and chip handlers, through recommendations of the department, set up a card file on all machines for draining and cleaning of coolants. The recommended drains and periodic cleaning were based on type of operation, type of metal being machined, and hours machines were run. One man makes periodic checks on pH and concentrations of the emulsions to supplement the draining and cleaning schedule. Since this system was put into operation, very few complaints have been experienced and there has been no rancidity to speak of.

The maintenance of water-soluble machining coolants in use is not difficult if a simple, but effective system of control is used. Supervision and other employees who use or handle these materials should be instructed by a qualified person. The instructions should include mixing, handling, check methods, good sanitary practices, and good housekeeping. Water-soluble machining coolants are doing a good job, and will do a better job if good control is used.

IN MEMORIUM. Mr. Harry A. Stevens Howarth, Research Engineer with the Franklin Institute and one of the early workers in the founding science of lubrication died in Philadelphia on April 21, 1955, at the age of seventy-eight. During his long and active career, the majority of which was devoted to the study of bearings and lubrication, he authored numerous valuable papers and remains today the major contributor to the subject of bearings and lubrication in Kent's **"Mechanical Engineers Handbook."** In December 1936, he was the recipient of the Melville Gold Medal in recognition of his work.

Mr. Howarth was born in New Haven, Connecticut, in 1877 and was graduated from Yale University in 1899 with honors in mechanical engineering. Several years of practical experience with steel mills and manufacturing were followed by teaching mechanical engineering subjects at Lehigh University and Carnegie Institute of Technology. In 1914 Mr. Howarth became associated with Albert Kingsbury who was then developing his famed pivoted pad thrust bearing and in 1917 became general manager and chief engineer. He was responsible for many of the major developments in the oil film bearings manufactured by this company and was awarded patents on about fifteen inventions in this field. He continued in the same position when Kingsbury founded the Kingsbury Machine Works in 1921. In 1928 he became vice-president and general manager.

In 1939, with M. J. Howarth he formed the Howarth Pivoted Bearings Company which manufactured a line of improved oil film type journal bearings and pivoted segment thrust bearings for industry, marine and naval applications until 1943.

Mr. Howarth was a bearing consultant and designer for the Bureau of Ships for three years on problems such as oil film bearings, thrust bearings and engine crankshaft bearings. In 1946 he became associated with the Franklin Institute as a research engineer where he was in charge of the design of various major components of ordnance gun mounts and carriages, and served as consultant on all bearing problems.

He was a member of the honorary society, Sigma Xi, and also of the American Society of Mechanical Engineers, and was a Fellow of the American Association for the Advancement of Science.

Mr. Howarth will always be held in high esteem by his fellow workers in the lubrication field for his pioneering analytical achievements and practical contributions to the science of lubrication. (J. Boyd & A. A. Raimondi)

ASLE Officers & Directors

ASLE Officers & Directors for 1955-1956. The following Officers & Directors were elected at the ASLE 10th Annual Meeting & Exhibit held at the Hotel Sherman, Chicago, Illinois, on April 13-14-15, 1955, to serve the American Society of Lubrication Engineers during the 1955-1956 term of office.

V -Pres. at Large
W. Deutsch
Trabon Engrg. Co.

Secretary
E. R. Booser
General Electric Co.

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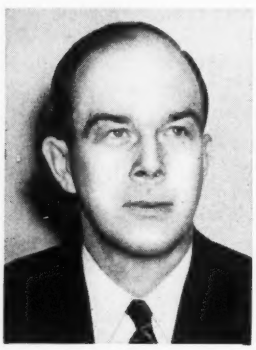

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Oil-Soaked Felt-Pad Lubrication Of Ball Bearings At High Speed & High Temperature

by H. S. White, J. F. Swindells & Harriet V. Belcher*

Conventional greases do not give satisfactory service in prepacked ball bearings at high speeds and high temperatures. At 325 F. and 10,000 rpm, endurance tests have been made with oil-soaked felt-pads which contact the outer race and bleed oil slowly to the bearing. With mineral oils, silicone fluids, and polypropylene glycol derivatives, there is a large variation in the length of the endurance runs, with some runs exceptionally long and check runs relatively short. With di(2-ethylhexyl) sebacate containing about 5% tricresyl phosphate and other additives the length of the tests before failure ranges from 1,005 to 1,781 hours.

In many devices it is desirable to use prelubricated ball bearings which require no further lubrication during the normal life of the bearings. Ball bearings which have been prepacked with grease are used in many applications; but these are not entirely satisfactory, especially where speeds and temperatures are high. In aircraft equipment it is desirable to have prelubricated ball bearings which will operate satisfactorily over the temperature range from —65 F. to +400 F. Conventional greases do not meet this requirement, chiefly because of excessive starting friction at the low temperatures and poor stability at the high temperatures.

In 1944 an apparatus was developed for the determination of the frictional torque of ball bearings operating over a wide range of speed at high temperatures. In one phase of this work, endurance runs were made with several greases in ball bearings operating at 10,000 rpm. These data for typical greases, at 300 F. and 325 F., are included in this paper for comparative purposes.

During operation at high speeds and high temperatures, the greases which gave the better performance were those which adhered to the outer race after being thrown from the path of the balls, and then lubricated the bearing by slow bleeding of the oil. Felt rings, saturated with oil, were then tried for bleeding oil to the bearings; essentially, felt was used instead of a soap or other thickener to supply oil slowly to the bearing, without further addition of oil to the felt. The data obtained during several years of endurance testing with this type of lubrication are presented in this paper to show the feasibility of using oil-soaked pads for prelubricated ball bearings.

*National Bureau of Standards, Engines & Lubrication Section, Washington, D. C.

APPARATUS. The apparatus uses a No. 204 (SAE No. 20BC02) ball bearing with a 20-mm bore as the test bearing. This is mounted out-board on a spindle which is supported by two bronze journal bearings that are lubricated by sight-feed oil cups. The outer race of the test bearing is mounted in a cylindrical housing provided with removable bearing shields at each end. An auxiliary sleeve around the housing increases the radial load to 1500 grams.

A pendulum attached to the under side of the housing is used in conjunction with a calibrated scale for measuring the frictional torque. The lower end of the pendulum dips into a damping fluid in an open pan, for minimizing fluctuations.

Two electric heaters are mounted below the housing. The heaters and bearing housing assembly are enclosed in an aluminum chamber. The temperature is controlled by a bimetallic regulator, and measured by a thermocouple contacting the outer race of the bearing and another in the air

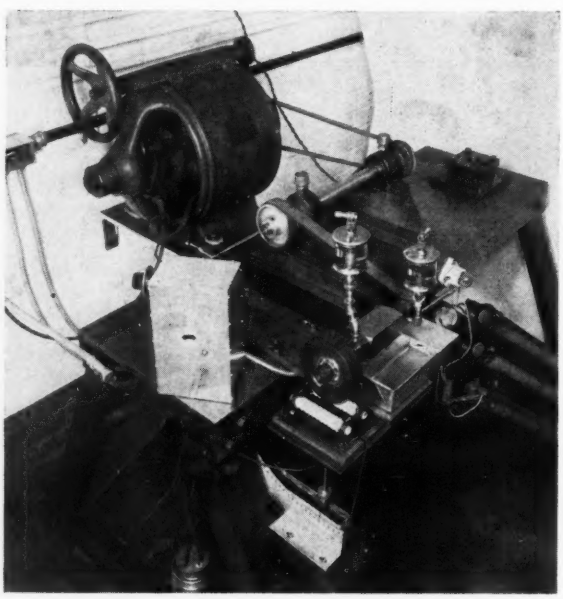

Fig. 1. High-temperature ball bearing testing machine with original variable-speed drive; with air chamber and bearing shield removed to show details of heaters and bearing housing.

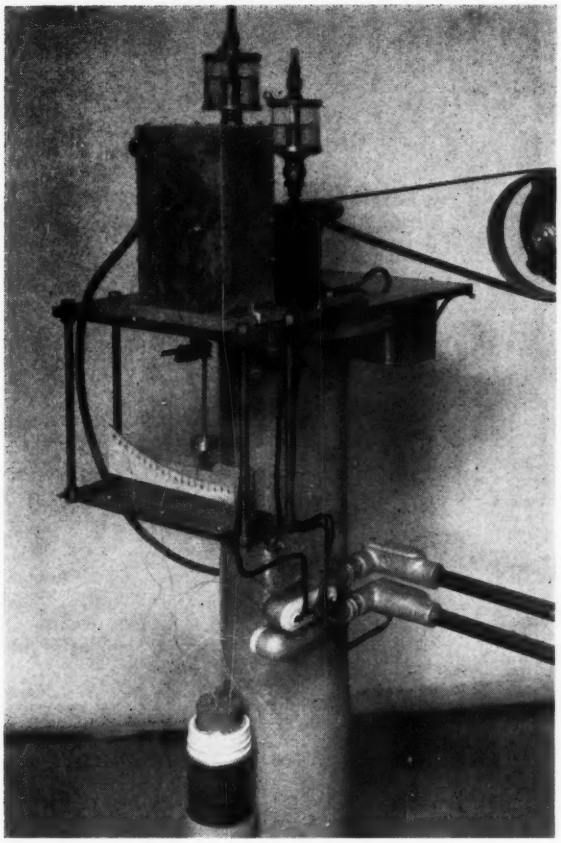

Fig. 2. High-temperature ball bearing testing machine, assembled, with 10,000 rpm drive.

space inside the chamber.

In the original apparatus, the spindle was driven by a variable-speed motor, through a countershaft. This apparatus is shown in Fig. 1, with the air chamber and front bearing shield not assembled.

For the endurance tests at 10,000 rpm the ball bearing testing machine was mounted on a pedestal and driven directly through a flat high-speed belt by a small motor. An automatic cut-off switch was added so that abnormally high torque would break the circuit to the motor and the heaters. Later, a duplicate of this machine was built, so as to have two machines for the endurance tests. One of these machines, assembled and in operation, is shown in Fig. 2.

In the preliminary tests with oil-soaked felt pads, bearing shields with annular grooves were fitted with the felt pads as shown in part A of Fig. 3. However, for the test data given in this paper, the felt pads were mounted in grooves in lightweight ring-shaped holders which were mounted in annual grooves provided in the bearing shields, as shown in part B of Fig. 3.

MATERIALS. Greases. A description of the greases used for comparative data on endurance runs with prelubricated bearings is given in Table I. The properties listed in this table were furnished by the suppliers of the samples.

Oils. Table II describes the oils used for saturating the felt pads in this type of prelubrication. The mineral oils ranged in viscosity from a transformer oil to an SAE No. 60 motor oil. Some were straight mineral oils; others were heavy duty type containing oxidation inhibitors and detergents.

The synthetic oils (SO series in Table II) include silicone fluids, polypropylene glycol derivatives containing oxidation inhibitors, and sebacates containing additives. In general, the samples of di(2-ethylhexyl) sebacate containing additives have pour points below —75 F. and flash points above 400 F.

Felts. The felts used in this investigation were SAE Nos. F-1, F-5, and F-10. No. F-1 is a relatively hard felt designated by the trade as "back check". No. F-5 has more resilience than F-1, and is designated as "extra firm pad"; No. F-10 is softer than F-5 and is designated as "firm pad" by the trade. Further description of these felts is given in SAE Handbooks.

TEST PROCEDURE. Grease Test Procedure. The ball bearings were cleaned with solvents and dried. For each test, a new bearing was packed with 3 g of the test grease and installed in position as shown in Fig. 1, but with the front bearing shield and the heating chamber in place as shown in Fig. 2. After turning on the heaters and starting the machine, the air chamber was heated sufficiently to bring the bearing outer race to the test temperature (300 F. or 325 F.) within the first half hour of operation.

In general, operation was continued with 21 hours of running and 3 hours of shutdown each day, except that at week ends the operation was for about 28 hours followed by shutdown over the week end. Each test was continued until failure of the bearing, as evidenced by excessive torque which operated the automatic overload device.

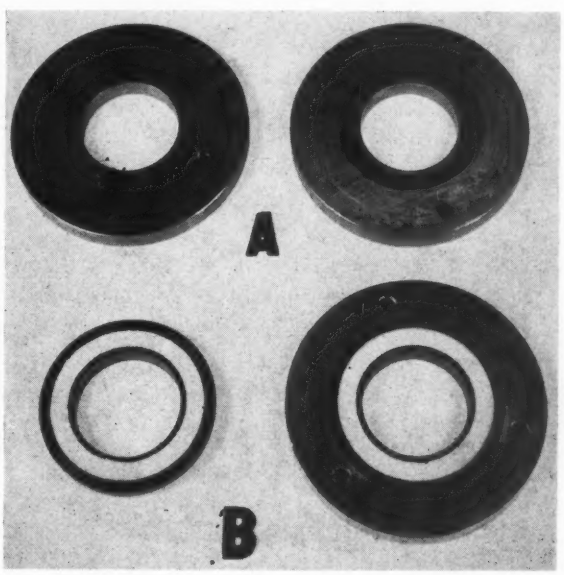

Fig. 3. Felt pads in holders and in bearing shields.

Oil-Soaked Felt-Pad Test Procedure. The ring-shaped felt pads were made from ¼-in. thick felt by means of concentric punches which gave the desired inner and outer diameters for the pads. The pads were mounted in the annular grooves of the ring-shaped holders, as shown in Fig. 3. Then they were immersed in the test oil at 340 F. for a few minutes until bubbling ceased, removed, and allowed to drain. This treatment at high temperature was necessary to remove air, moisture, and other volatile matter from the felt; otherwise, bubbling tended to remove most of the oil from the felt in the ball bearing assembly during the first heating to 325 F.

For each test, a new ball bearing was cleaned with solvents, dried, dipped in the test oil at 340 F., and allowed to drain. The ball bearing was installed in a manner similar to that described for the greased bearings, with an oil-soaked pad on each side. The design of the felt pads and their installation is such that the felt contacts the outer race but does not contact the inner race and ball separators of the ball bearing. Each test was continued until failure of the ball bearing, in the same manner as that described for the greased bearings.

TEST RESULTS & DISCUSSION. The data were obtained with Conrad or non-filling slot type ball bearings, each containing 8 steel balls retained by steel separators. These are known as No. 204 single row radial ball bearings (SAE No. 20BC02). Most of the data were obtained with standard bearings; but 6 tests were made with heat-stabilized bearings (designated HS in Table IV), and 2 tests were made with specially selected bearings from another manufacturer.

Greases. The endurance data for the ball bearings prepacked with test greases are given in Table III. These are presented as typical data for greases for comparison with the data obtained with oil-soaked felt-pad lubrication. At 300 F. and 10,000 rpm, 3 greases failed in less than 100 hours average time; 2 of the greases provided operation over 400 hours. Sample G-6, very similar to G-4, (see Table I) is a typical grease used for the lubrication of ball bearings at high temperatures; at 325 F. and 10,000 rpm the length of tests before failure ranged from 178 to 330 hours.

Mineral Oils. The endurance data for the ball bearings lubricated with felt pads soaked in mineral oils are given in the upper section (MO series) of

Table I. Properties of grease samples.

Sample Code No.	ASTM Penetration at 77°F		Soap		ASTM Dropping Point, °F	Water, percent	Mineral Oil Visc., centistokes	
	Unworked	Worked	Percent	Type			at 100°F	at 210°F
G-1	240	293	10.6	Na	342	Trace	52.7	6.2
G-2	220	270	15.5	Na	342	0.5	22.0	4.0
G-3	275	285	20.0	Na-Ca	360	Trace	43.2	5.7
G-4	275	290	23.5	Na	450+	Nil	67.1	6.7
G-5	268	278	23.5	Na	436	Trace	242.	19.3
G-6	254	275	26.	Na	400	--	67.1	--

Table II. Properties of oil samples.

Sample Code No.	Type of Oil	Visc. centistokes		Visc. Index	Oxidation Inhibitor
		at 100°F	at 210°F		
MO-1	Transformer oil	8.9	2.3	71	No
MO-2	SAE 20 (1942 supply)	83	9.9	107	Yes
MO-3	SAE 20 (1947 supply)	81	9.7	106	Yes
MO-4	SAE 30	112	12.0	103	Yes
MO-5	SAE 30	112	11.3	95	Yes
MO-6	SAE 30	121	--	--	Yes
MO-7	SAE 30	124	--	--	Yes
MO-8	SAE 40	149	14.7	106	--
MO-9	SAE 60	330	25	104	No
SO-1	Dimethyl silicone	74	30	156	No
SO-2	Medium phenyl-methyl silicone	95	21	152	No
SO-3	High phenyl-methyl silicone	131	22	143	No
SO-4	Experimental silicone	82	--	--	--
SO-5	Polypropylene glycol derivative	119	18.3	139	Yes
SO-6	" " "	121	--	--	Yes
SO-7	" " "	113	--	--	Yes
SO-8	" " "	124	--	--	Yes
SO-9	Di (tetradecyl) sebacate	44	--	--	Yes
SO-10	Di (2-ethylhexyl) sebacate	12.7	3.3	149	Yes
SO-11	SO-10 + 5% TCP	12.7	3.3	149	Yes
SO-12	SO-11 + 4% VI improver	22.5	5.6	176	Yes
SO-13	SO-10 + TCP + other additive	14.0	3.6	162	Yes
SO-14	SO-12 + 4% VI improver + 0.5% acid phosphate	43	10	163	Yes

Table IV. As would be expected, the relatively rapid evaporation of the transformer oil (MO-1) at 325 F. resulted in failure in about 26 hours.

With the sample of SAE No. 20 motor oil obtained in 1942 (MO-2), using standard bearings, in 5 tests the results range from failure at 162 hours to no failure at 1,000 hours. In view of this wide variation in data obtained with the standard bearings, it is believed that additional tests with the selected and heat-stabilized bearings would be necessary before attaching much significance to the longer average length of the few tests with these bearings. This same reasoning applies to the tests with other oils; the great length of time necessary to complete a test of about 2,000 hours did not permit obtaining the desired number of check runs. However, the trend shows no advantage in using an oil higher in viscosity than the SAE 20 oil.

The sample MO-2 of SAE 20 oil is described by the manufacturer as containing a relatively small amount of an additive which acts as an extreme-pressure and oiliness agent and as an anti-oxidant and bearing corrosion inhibitor. Sample MO-3 was obtained from the same manufacturer 5 years later and described as containing about 1 percent of a detergent and bearing corrosion inhibitor. Samples MO-8 (SAE 40) and MO-9 (SAE 60) were obtained from the same manufacturer as the above samples of SAE 20; it is assumed that sample MO-9 contains no additive since it was purchased as meeting the requirements for a Navy Symbol 1120 oil. It is possible that the additives present in the SAE 20 oil may have been responsible for its performance being somewhat better than that of the SAE 60 without additives.

Synthetic Oils. The endurance data for the ball bearings lubricated with felt pads soaked in synthetic oils are given in the lower section (SO series) of Table IV. The average length of the endurance tests with the 4 silicone fluids (SO-1 to SO-4) is somewhat greater than the average for the mineral oils. With the silicone fluids, the duration of the endurance tests ranges from 85 to 2,309 hours.

With the 4 samples (SO-5 to SO-8) of polypropylene glycols containing different oxidation inhibitors and the sample of inhibited di(tetra-decyl) sebacate, the average length of the endurance tests are relatively short.

With di(2-ethylhexyl) sebacate containing no additive except an oxidation inhibitor the length of

Table III. Test data for grease samples, with Standard No. 204 ball bearings, at 10,000 rpm.

Sample Code No.	Bearing Temp., °F	Hours Run before Failure			
		Run 1	Run 2	Run 3	Avg
G-1	300	11	40	--	26
G-2	300	18	44	--	31
G-3	300	93	103	--	98
G-4	300	404	--	--	404
G-5	300	546	--	--	546
G-6	325	178	307	330	272

Table IV. Test data for wick-fed oil samples, with No. 204 ball bearings, at 10,000 rpm, at 325 F.

Sample Code No.	Bearing	Felt, SAE No.	Hours Run before Failure					
			Run 1	Run 2	Tun 3	Run 4	Run 5	Avg
MO-1	Std	F-1	23	28	--	--	--	26
MO-2	Std	F-1	396	1000+	329	202	162	418+
MO-2	Select	F-1	541	375	--	--	--	458
MO-2	HS	F-1	2008	--	--	--	--	2008
MO-3	Std	F-1	2243	--	--	--	--	2243
MO-4	Std	F-1	399	--	--	--	--	399
MO-5	Std	F-1	749	--	--	--	--	749
MO-6	HS	F-1	146	182	--	--	--	164
MO-7	HS	F-1	1270	--	--	--	--	1270
MO-8	Std	F-1	321	--	--	--	--	321
MO-9	Std	F-5	675	182	--	--	--	428
MO-9	Std	F-1	172	--	--	--	--	172
SO-1	Std	F-1	914	85	--	--	--	500
SO-2	Std	F-1	2309	782	--	--	--	1546
SO-3	Std	F-1	1348	268	--	--	--	808
SO-4	HS	F-1	1024	--	--	--	--	1024
SO-5	Std	F-5	114	12	81	--	--	69
SO-5	Std	F-10	113	--	--	--	--	113
SO-6	Std	F-5	237	297	--	--	--	267
SO-6	Select	F-5	169	294	--	--	--	232
SO-6	HS	F-5	223	--	--	--	--	223
SO-7	Std	F-5	94	186	--	--	--	140
SO-8	Std	F-5	262	383	--	--	--	322
SO-9	Std	F-1	245	--	--	--	--	245
SO-9	HS	F-1	325	--	--	--	--	325
SO-10	Std	F-1	48	--	--	--	--	48
SO-11	Std	F-1	1005	--	--	--	--	1005
SO-12	Std	F-1	1246	--	--	--	--	1246
SO-13	Std	F-1	1498	--	--	--	--	1498
SO-14	Std	F-1	1781	--	--	--	-	1781

the endurance test was only 48 hours. However, with the 4 samples (SO-11 to SO-14) of this diester containing tricresyl phosphate the length of test before failure was over 1,000 hours. Sample SO-11 is the reference oil given on page 3 of Military Specification MIL-L-7808. Samples SO-12 to SO-14 are essentially this reference oil with other additives designed to increase its load-carrying performance at high temperatures. The average length of tests before failure with the 4 samples of the diester containing tricresyl phosphate is greater than for any of the other types of oil tested. Although the number of tests are too few to be conclusive, the addition of the viscosity index improver to sample SO-11 apparently increased the length of test from 1,005 hours to 1,246 hours, and the further addition of a mild extreme-pressure additive (acid phosphate) increased the length of test to 1,781 hours. These data are consistent with the expected effect of the additives.

Comparison & Discussion. The number of check tests performed are not sufficient to justify the formulation of statistical conclusions for the data presented. However in Table III for the typical greases tested at 300 F. and 10,000 rpm, the mean of the averages is 221 hours. In Table IV for felt-pad lubrication at 325 F. and 10,000 rpm, the mean of the averages for the mineral oils ranging from SAE 20 to SAE 60 grade is 785 hours (neglecting the plus resulting from a run that was stopped at 1,000 hours); 970 hours for the 4 silicone fluids; 195 hours for the glycol derivatives; and 1,382 hours for the 4 samples of di(2-ethylhexyl) sebacate containing tricresyl phosphate. The following minimum values obtained at 325 F. may be more significant than the above means: high temperature grease, 178 hours; SAE 20 motor oil, 162 hours; medium phenyl-methyl silicone, 782 hours; glycol derivative sample SO-6, 169 hours; sebacate containing tricresyl phosphate, 1,005 hours.

Of the various types of oils tested it is believed that the di(2-ethylhexyl) sebacate containing tricresyl phosphate and other appropriate additives is the most promising. In addition to its excellent performance at 325 F., its viscosity characteristics at low temperatures indicate that it would be the most suitable type for low-torque starting at −65 F.

The wool felts used in these tests became charrred at 325 F., especially after operating over 1,000 hours. For operation at temperatures above 325 F. it is possible that pads made from other fibers, such as metal, glass, or polytetrafluoroethylene, might be suitable.

CONCLUSIONS. Although not conclusive, the following trends are indicated from the data.

1. Under the conditions used in these tests, when suitable fluids are used, the oil-soaked felt-pad method of lubricating ball bearings shows some promise of being superior to the practice of prepacking with grease.

2. A low pour point mineral oil, such as transformer oil, is not suitable for operation at 325 F. because of its high volatility at that temperature.

3. There appears to be no advantage in using an oil higher in viscosity than an SAE 20 grade, for the lubrication of ball bearings at high speed and high temperature.

4. Di(2-ethylhexyl) sebacate containing suitable additives shows promise for providing satisfactory long-time lubrication at 325 F.

5. Additives of the mild extreme-pressure antiwear type, such as tricresyl phosphate, appear to be very effective when combined with di(2-ethylhexyl) sebacate for the long-time lubrication of ball bearings at high temperature.

ACKNOWLEDGEMENT. This work was initiated and supervised by Mr. S. A. McKee, former chief of the Engines & Lubrication Section at the National Bureau of Standards, who retired in January of 1953.

New Products

Dromus Oil E, a newly-developed synthetic cutting fluid, when dissolved in water has been perfected to yield a safe and economical cutting solution for use in all types of machine tools. Since it actually dissolves in water (forming a true solution instead of an emulsion), even on long standing no separation is noticeable; the solution is stable at all temperatures normally encountered. Since it may be mixed with a large volume of water to utilize the latter's very high cooling effect, higher speeds and feeds are thus possible. The addition of Dromus Oil E prevents rusting of parts or equipment; it also improves the wetting characteristics of the solution to reduce friction between tool and chip, and to increase cooling still further. A properly prepared solution will not froth or foam objectionably, or form hard-to-clean sludge. Chips and grinding grits settle out rapidly so that most of these are not recirculated through pumps and cooling systems. If the machine is to be cleaned or dismantled, the coolant may be syphoned off the top and reused. For metal cutting machinery such as lathes, boring mills, drill stands, milling machines, and the like, a solution of 30 parts of water to 1 part Dromus Oil E is recommended. Depending on plant experience, this ratio may be changed to as high as 40:1 or as low as 20:1. (Shell Oil Co., LE-11/3, 50 W. 50th St., NYC 20.)

Valley Barrel Cart. A new barrel handling truck, Model 600, has been announced that requires no tipping or pick-up space when handling barrels touching each other. Because of an automatic spring actuated bar that allows a hook to slip over the barrel edge as the truck is moved against the barrel (without the operator

(Continued on p. 203)

Recent Advances In The Hydrodynamic Theory Of Slider-Bearing Lubrication

by F. Osterle & E. Saibel*

This paper is a survey of some recent contributions to the hydrodynamic theory of lubrication made by the present authors and their co-workers. The momentum (Reynolds) equation, governing the isothermal lubrication of slider bearings with side flow, is solved exactly for rectangular or sector sliders with exponential profiles. The results are extended to include the effect of variable lubricant viscosity, transverse slider curvature, and certain nonsteady-state phenomena. The energy equation governing the adiabatic lubrication of slider bearings is examined and simultaneous solutions of this equation, and the appropriate momentum equation are obtained for both inclined and parallel slider bearings. The problem of the optimum slider profile for isothermal flow of the lubricant is reconsidered under more realistic conditions, and the performance of the stepped profile slider with adiabatic flow is investigated. The work described in this paper is abstracted from a series of nine articles, published in the Transactions of The American Society of Mechanical Engineers, under the general title "On the Solution of the Reynolds Equation for Slider-Bearing Lubrication."[1]

THE REYNOLDS EQUATION. The fundamental problem in the hydrodynamic theory of slider-bearing lubrication is the prediction of the pressure distribution in the lubricant film between two parallel or nearly parallel surfaces, very close together, and in relative tangential motion. This pressure distribution is governed in part by the so-called Reynolds equation which is derivable from the Navier-Stokes and continuity equations of hydrodynamics. This equation can be written

$$\frac{\partial}{\partial x}\ (\rho\, q_x) + \frac{\partial}{\partial y}\ (\rho\, q_y) = 0$$

with

$$q_x = \frac{Uh}{2} - \frac{h^3}{12\mu}\frac{\partial p}{\partial x}, \quad q_y = \frac{h^3}{12\mu}\frac{\partial p}{\partial y} \tag{1}$$

where q_x = lubricant flow in x-direction per unit length in y-direction; q_y = lubricant flow in y-direction per unit length in x-direction; ρ = lubricant density; p = lubricant pressure; μ = lubricant viscosity; h = film thickness; U = velocity of moving surface (bearing) in x-direction, Fig. 1. The boundary condition generally employed with the Reynolds equation is that the pressure at the perimeter of the slider is some constant value.

Implicit in the Reynolds equation are the assumptions that the inertia and weight of the lubricant are negligible and that the film is so thin that the pressure, density, and viscosity of the lubricant are essentially constant throughout its thickness. That the density and viscosity are not constant throughout the film length and width is apparent from the fact that as the lubricant flows between the slider and bearing its temperature rises due to viscous friction and both density and viscosity vary with temperature. Furthermore the viscosity of the lubricant is appreciably affected by pressure. It will be shown later that with the viscosity considered a function of pressure the problem is still governed completely by the Reynolds equation and the appropriate boundary conditions, but if temperature effects are to be taken into account an energy equation is required in addition to the Reynolds equation to govern the problem.

In most of the early work in the theory of lubrication the lubricant was considered incompressible and the viscosity was taken as a constant, presumably at some mean value. Solutions to Equation (1) with side flow neglected were obtained easily. Equation (1) with side flow considered was solved first by Michell[2] for a rectangular slider with a linear profile. It is well known that this solution

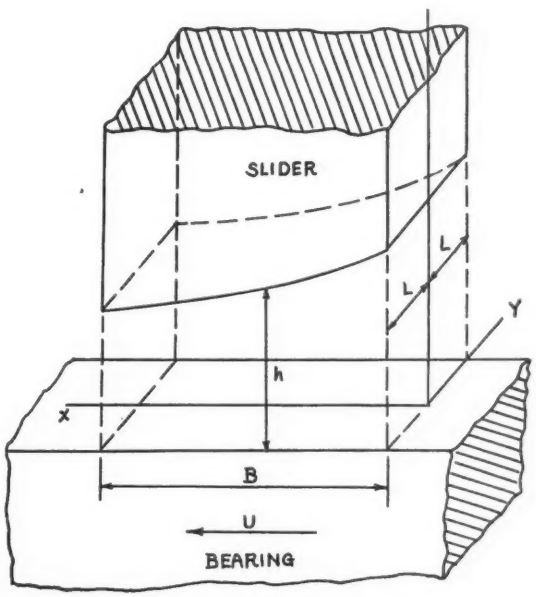

Fig. 1.

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(This paper was contributed by ASLE, and presented at the 1st Annual ASME-ASLE Lubrication Conference, Baltimore, Oct. 18-20, 1954.)

is extremely difficult to apply to practical problems for which reason numerous approximate methods have been devised to obtain numerical results. Most of these are useless for the general case of any slider length-width ratio or any profile inclination.

THE RECTANGULAR SLIDER WITH EXPONENTIAL PROFILE. In Part I of Ref. 1 it was found that for a rectangular slider with a profile such that the film thickness is given by

$$h = h_m\, r^{-(1-x/B)} \tag{2}$$

where h_m is the minimum (outlet) film thickness and r is the outlet to inlet film-thickness ratio, Equation (1) becomes quite tractable and the pressure distribution for any L/B ratio and any r value is easily obtained in the form of a rapidly convergent series from which calculations for the load-supporting capacity, center of pressure, and friction force can be made quickly.

The exponential profile does not differ very much from linear being only slightly convex. Interestingly enough, Raimondi and Boyd[3] point out that the usual lapping or scraping techniques used in the manufacture of plane sliders will produce pads in which the departure from flatness usually will be in the direction of making the pad surface convex. It will be seen later that there is very little difference between the load-capacity of the exponential slider and the plane pad.

The solution of Equation (1) upon substitution of Equation (2) can be written as the sum of two terms, P which is the pressure distribution which would result if side flow were neglected, in which case the second term in Equation (1) would vanish, and p' which can be thought of as a "correction" term to account for side flow. It is easily shown that P takes the form

$$P = \frac{6\mu UB}{h_m^2}\,\frac{r^2-1}{2\ln r}\left(\frac{1-r^{3x/B}}{1-r^3}-\frac{1-r^{2x/B}}{1-r^2}\right) \tag{3}$$

which can be integrated over the slider length to give the average pressure

$$\overline{P} = \frac{6\mu UB}{h_m^2}\,\frac{r^2-1}{2\ln r}\left(\frac{1-(r^3-1)/3\ln r}{1-r^3} - \frac{1-(r^2-1)/2\ln r}{1-r^2}\right) \tag{4}$$

In Table I, below, the results of Equation (4) are compared with those obtained for the plane slider at several values of r to show the smallness of the deviation

r	$\frac{\overline{P}h^2_m}{6\mu UB}$ (*exponential*)	$\frac{\overline{P}h^2_m}{6\mu UB}$ (*linear*)
1/2	0.0267	0.0264
1/3	0.0253	0.0246

With side flow considered, the term $\overline{p}'$ (which works out to be a convergent series) must be added to $\overline{P}$ as given by Equation (4). The resulting total average pressure $\overline{p}$ for a square slider with an r value of 1/2 works out to be

$$\overline{p} = 0.0113\,\frac{6\mu\, UB}{h_m^2}$$

Michell's solution for the plane slider yields

$$\overline{p} = 0.0115\ 6\mu UB/h_m^2$$

The difference between these solutions is less than 2 per cent. Comparing these results with Table I shows to what extent side-flow reduces the load capacity.

EFFECT OF TRANSVERSE CURVATURE. In Part III of Ref. 1 the effect of transverse curvature on the properties of a rectangular slider with exponential profile is investigated. Film-thickness variations of the form

$$h(x,y) = h\,(x)\ s^{(y/L)^2} \tag{5}$$

where L is the half-width of the slider (Fig. 1) and s is the side to center film-thickness ratio, are considered. If we let s equal $(1 + c)$ Equation (5) can be written as a power series in c

$$h(x,y) = h\,(x)\left[1 + c\,(y/L)^2 + \cdots\right] \tag{6}$$

The pressure distribution corresponding to this film-thickness variation also can be written as a power series in c.

$$p = p_o + c\,p_1 + \cdots \tag{7}$$

where p_o, p_1, $\cdots$ can be determined by substituting Equations (6) and (7) into the Reynolds Equation (1), equating like powers of c and solving the resulting equations. The function p_o will of course be the solution corresponding to no transverse curvature. If c is sufficiently small, Equation (7) becomes

$$p = p_o + c\,p_1 \tag{8}$$

and corresponding to this the average pressure is given by

$$\overline{p} = \overline{p}_o + c\,\overline{p}_1 \tag{9}$$

and the friction force by

$$F = F_o + c\,F_1 \tag{10}$$

The second terms in Equations (8), (9), and (10) are thus corrections to account for transverse curvature.

For a square slider with an r value of 1/2 we previously showed that

$$\overline{p}_o = 0.0113\ 6\mu UB/h_m^2$$

and in Part I of Ref. 1 it is shown that

$$F_o = 0.1259\ 6\mu UB^2/h_m$$

The terms $\overline{p}_1$ and F_1 work out to be

$$\overline{p}_1 = -0.018\ 6\mu UB/h_m^2$$

$$F_1 = -0.0445\ 6\mu UB^2/h_m$$

Therefore for sufficiently small c

$$\overline{p} = (0.0113 - 0.0118\,c)\ 6\mu UB/h_m^2$$

and

$$F = (0.1259 - 0.0445\,c)\ 6\mu UB^2/h_m$$

Thus if the slide film thickness is for example 1 per cent greater (smaller) than the center film thickness the average pressure is reduced (in-

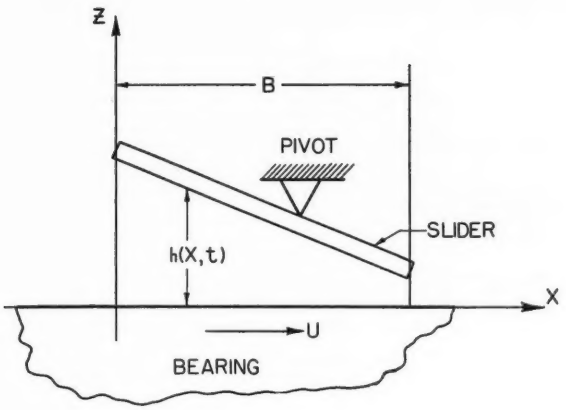

Fig. 2.

creased) by about 1 per cent and the friction force is reduced (increased) by about 1/3 per cent. In other words convex curvature decreases the average pressure and friction force while concave curvature increases both.

NONSTEADY-STATE OPERATION. It has long been realized that the load-supporting capacity and coefficient of friction exhibited by bearings under nonsteady-state operating conditions differ markedly from the predictions of steady-state theory. For example, Gatcombe[4] conducted experiments on oil films entrapped between the peripheral surfaces of rotating disks and found that under transverse vibration of the disks the load supported by the film could be as much as 6 times as large as the load predicted by steady-state theory. As a contribution to the understanding of the nonsteady-state operation of bearings a specific nonsteady-state problem is worked out in Part VII of Ref. 1. This problem is the tilting-pad slider bearing, Fig. 2, subject to a sinusoidal pivot vibration in the direction normal to the bearing surface.

The results of this investigation can be summarized as follows:

(a) For the pressure in the film to be positive at all times the amplitude (a) and frequency (ω) of the impressed sinusoidal oscillation must satisfy the following inequality

$$\Omega < (1 - A^2)^{1/2}/A \qquad (11)$$

where

$$\Omega = -2\omega B/U \ln r, \text{ and } A = a_1/h_1^*$$

with a_1 the amplitude of the pivot oscillation and h_1^* the mean value of the film thickness at the pivot. The other variables are as defined previously. The dashed line in Fig. 3 is so placed that all points to the left of it satisfy this inequality hence represent states of positive film pressure.

(b) The mean load-supporting capacity ($\overline{W}$) increases with the amplitude of oscillation but is independent of oscillation frequency. This relationship is given by

$$\overline{W}/W^* = 1/(1 - A^2)^{1.5} \qquad (12)$$

where W^* is the load capacity under steady-state operation. Equation (12) is plotted in Fig. 3.

(c) The mean coefficient of friction increases with both the amplitude and frequency of oscillation approaching an infinite value as the film pressure approaches zero.

SECTOR SLIDER WITH EXPONENTIAL PROFILE. No analytical solution exists at the present time for the problem of the plane-sector slider bearing with side flow considered. The problem can be solved by relaxation methods as Brand[5] has done. In Part V of Ref. 1 a sector slider is considered having a profile such that the film thickness is given by

$$h = h_m r^{-\theta/\beta} \qquad (13)$$

where h_m is the minimum (outlet) film thickness, r is the outlet to inlet film-thickness ratio, θ is the polar angle measured from the outlet edge of the pad, and β is the sector angle. The pressure distribution is easily obtained as in the case of the rectangular slider.

The extent to which the sector slider with exponential profile approximates the sector slider with linear profile was determined by comparison with the work of Brand. Brand considers a plane sector pad tilted at an angle α around a radial line bisecting the sector angle β and parallel to the bearing surface but separated from it by a distance h_o. A comparison with his results is made by letting his h_o be the mean value of the inlet and outlet film thicknesses of Equation (13) and letting his value of α be the difference between the inlet and outlet film thicknesses divided by the chord distance between the mid-point on the inlet edge of the slider and the mid-point on the outlet edge. It is found that the load-capacity and frictional moment agree extremely well with Brand's results for values of r greater than about 0.4.

EFFECT OF PRESSURE ON VISCOSITY. In Part II of Ref. 1 it is shown that the effect due to the variation of viscosity with pressure can be

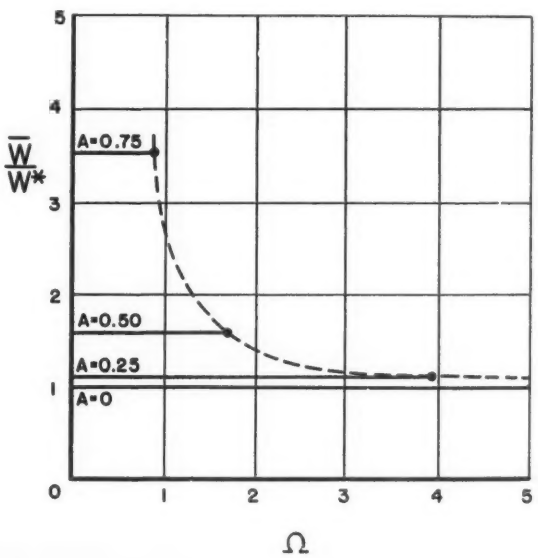

Fig. 3.

taken into account quite readily even with side flow considered.

If we let the zero pressure viscosity be given by μ_o, the well-known transformation

$$p^* = \mu_o \int_o^p \frac{dp}{\mu} \tag{14}$$

reduces the Reynolds Equation (1) to

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu_o}\frac{\partial p^*}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{h^3}{\mu_o}\frac{\partial p^*}{\partial y}\right) = 6U\frac{\partial h}{\partial x} \tag{15}$$

which is precisely the Reynolds equation which would apply if the viscosity was constant at μ_o, p^* being the pressure distribution for this situation.

If we introduce the following widely recognized viscosity-pressure relationship:

$$\mu = \mu_o e^{ap} \tag{16}$$

it follows from Equation (14) that

$$p = -(1/a)\ ln\ (1 - ap^*) \tag{17}$$

Thus once p^* is determined by the methods previously described it can be converted into the actual film pressure by Equation (17).

If the series expansion for the logarithm is applied to Equation (17) we have

$$p = p^* + \frac{a}{2} p^{*2} + \cdots \tag{18}$$

for ap^* less than unity, and since both a and p^* are nonnegative quantities it follows that at corresponding points in the film p is greater than p^*. In other words, the film pressure taking viscosity variation into account is greater than the film pressure neglecting this variation.

It can be shown that if ap is small compared to unity (which is usually the case), an approximation to the average pressure is given by

$$\bar{p} = \bar{p}^*\left(1 + \frac{a}{A}\ \bar{p}^*\right) \tag{19}$$

where $\bar{p}^*$ is the average pressure corresponding to p^* and A is the slider area. A typical value of a is 15×10^{-5} in^2/lb so that, for example, a predicted average pressure of 1000 psi would be about 4 per cent low for a 2-sq-in. slider if viscosity variation were neglected.

ENERGY EQUATION. In the solutions presented so far in this paper the viscosity has been assumed either constant or a function of pressure. Therefore they strictly apply only to the case in which the lubricant flows "isothermally" between the slider and bearing. It is well known, however, that except in unusual cases the temperature of the lubricant does not remain constant as it flows through the bearing but rises enough to cause the viscosity to decrease appreciably.

The temperature of the lubricant rises as internal energy is stored in it. The conservation-of-energy principle requires that the internal energy stored in the lubricant be equal to the work done on it by the viscous and pressure forces less the heat conducted away from it. Isothermal flow implies that all of the work done on the lubricant is conducted away as heat. However, there is evidence that in almost all cases most of the work done on the lubricant is stored in it as internal energy and not conducted away as heat. Therefore it would be important to consider the properties of slider bearings under conditions of "adiabatic" lubricant flow. The real situation would lie somewhere between the adiabatic and the isothermal bounds, presumably closer to the adiabatic bound.

For some time there had been disagreement in the literature over the proper expression to be used for the rate at which work is done on the lubricant. The present authors with A. Charnes[6] discovered that some investigators had omitted the work done by the pressure forces from their work-rate expressions. With this the so-called "flow-work" of thermodynamics considered the discrepancy was resolved.

Assuming that the internal energy of the lubricant remains constant throughout the film thickness, the energy equation for adiabatic flow can be written

$$S_x\ U = \rho\left(\ q_x\ \frac{\partial i}{\partial x} + q_y\ \frac{\partial i}{\partial y}\ \right) \tag{20}$$

with

$$S_x = \frac{\mu U}{h} + \frac{h\partial p}{2\partial x}\ and\ i \equiv e + \frac{p}{\rho}$$

where S_x = frictional force per unit film area exerted on lubricant by moving surface, e = lubricant internal energy, i = lubricant enthalpy.

The viscosity and internal energy of the lubricant are not constant but vary with pressure and temperature. Once these laws are known the energy equation and the momentum (Reynolds) equation reduce to simultaneous equations in pressure and temperature as the two unknowns.

An adequate representation of the variation of viscosity with pressure and temperature is shown in Part IV of Ref. 1 to be

$$\mu = \mu_o\ e^{ap}\ e^{-\gamma t} \tag{21}$$

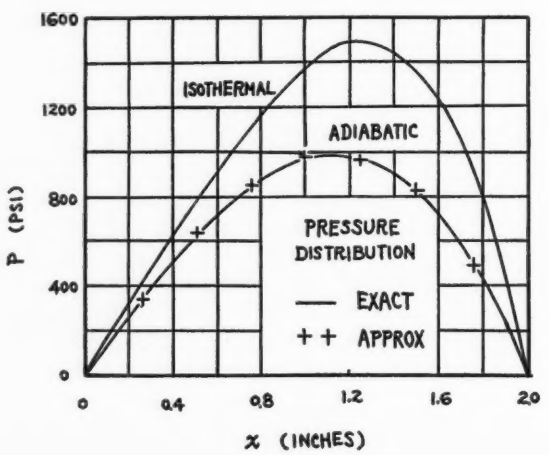

Fig. 4.

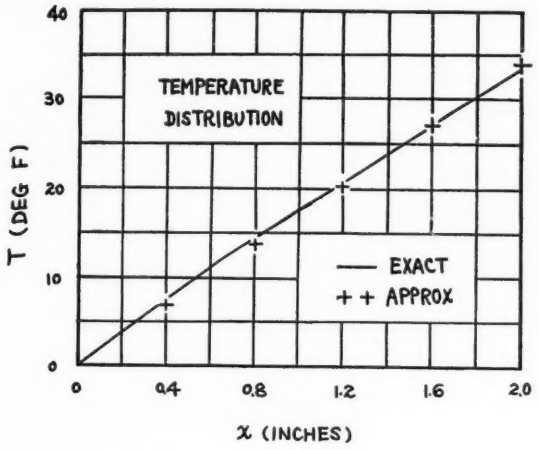

Fig. 5.

where μ_o is the viscosity of the lubricant at entry, p is the lubricant pressure measured above entry value, and t is the lubricant temperature measured above entry value.

The internal energy is assumed independent of pressure and linearly related to temperature

$$e = e_o + c\,t \tag{22}$$

where e_o is the internal energy of the lubricant at entry and c is its specific heat.

INCLINED SLIDER WITH ADIABATIC FLOW. In Part IV of Ref. 1 the momentum and energy equations as applied to the inclined slider are solved simultaneously for the case of no side flow by an exact method and for the case with or without side flow by an approximate method. The two methods are applied to the case of an inclined slider-bearing operating under the following conditions: bearing length = 2 in., bearing velocity = 320 ips, inlet film thickness = 0.002 in., outlet film thickness = 0.001 in. The lubricant is taken to be SAE 20 entering at 100 F. for which it is reasonable to assume a density of 0.032 pci, a specific heat of 4300 in-lb/lb-deg. F., a viscosity at inlet of 9.25 microreyns, an α value of 15×10^{-5} in^2/lb, and γ-value of 0.029 per deg F. The results of the calculations are shown in Figs. 4 and 5.

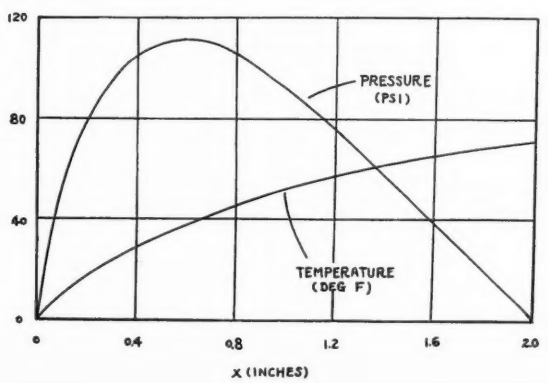

Fig. 6.

For comparison, the pressure distribution which would result if the flow was isothermal is also shown in Fig. 4.

PARALLEL SLIDER WITH ADIABATIC FLOW. Up to this point in the paper all the problems have been solved on the basis of an incompressible lubricant. Under this restriction zero load-capacity is predicted for the parallel-surface slider bearing, yet such bearings have been observed to support an appreciable load. In this case, apparently, the variation of density with temperature, although small, must be taken into account. In Part VI of Ref. 1 the momentum and energy equations are solved simultaneously for such a bearing with the density taken to vary with temperature according to the equation

$$\rho = \rho_o\, e^{-\lambda t} \tag{23}$$

where ρ_o is the density of the lubricant at entry. Side flow is ignored in this analysis.

The solution is applied to the case of a parallel slider bearing operating under the following conditions: bearing length = 2 in., bearing velocity = 800 ips, film thickness = 0.001 in. The lubricant is specified as a medium oil entering with a viscosity of 9 microreyns and a specific gravity of 0.90. It is further assumed that the lubricant has a specific heat of 4200 in-lb/lb deg F., a γ-value of 0.025 per deg F., and a λ-value of 4×10^{-4} per deg F. The α-value is taken to be the same as before. The results of the calculations are shown in Fig. 6. The load supported by this "density wedge" is characteristically much less than the load supported by the "geometric wedge" of the inclined slider, as Figs. 4 and 6 indicate.

THE STEPPED SLIDER. In his classical paper[7] Lord Rayleigh proposed and solved the following problem in hydrodynamic lubrication: For isothermal flow of the lubricant what slider profile will make the load capacity of a slider bearing a maximum subject to the conditions of a given slider length and a given minimum film thickness? In his solution Rayleigh did not permit variation of the lubricant viscosity with pressure.

The engineering importance of this optimum problem, especially in the design of heavily loaded

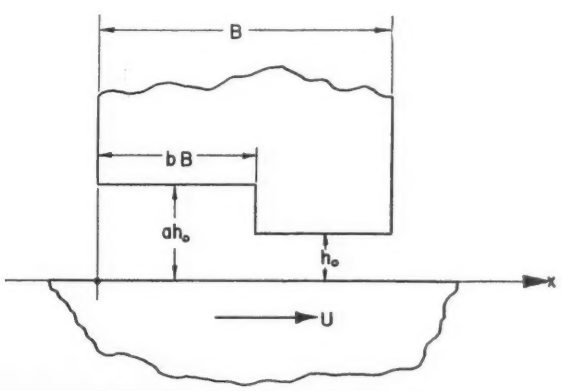

Fig. 7.

rubbing surfaces of machine elements, is obvious and it would thus seem worth while that the solution be sought which would take into account the variation of viscosity brought about by the high pressures associated with these heavy loads.

Such a solution is worked out in Part VIII of Ref. 1 and it is found that the optimum profile for variable viscosity is precisely the stepped profile which Rayleigh determined to be optimum for constant viscosity (Fig. 7 with $a = 1.8660$, and $b = 0.7182$). However the optimum load is greater. For example, consider a slider bearing with the optimum stepped profile operating under the following conditions: bearing length = 2 in., bearing velocity = 320 ips, minimum film thickness = 0.001 in. Further, consider the lubricant to be the SAE 20 oil described in the example concerned with the inclined slider bearing. If the viscosity variation is neglected (a la Rayleigh) the load capacity works out to be 2440 lb per in. of slider width. If the viscosity variation is taken into account the load becomes 2810 lb per in. of slider width, an increase of about 15 per cent.

In Part IX of Ref. 1 the stepped-profile slider bearing just discussed is analyzed under conditions of adiabatic rather than isothermal lubricant flow. For this case the load capacity works out to be 1680 lb per in. of slider width, 31 per cent less than the Rayleigh case.

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Personals

Dr. M. E. Merchant (ASLE Past-President), Cincinnati Milling Machine Co., was presented the 6th Annual Award of the Technical & Scientific Societies Council of Cincinnati, as **Outstanding Engineer of the Year.** (The selection of the recipient of this award is based upon the following three aspects of individual achievement: His contribution (1) to scientific, engineering, or technical knowledge; (2) to the development and well being of the profession; and (3) to human welfare.)

Lincoln Engineering Co., St. Louis, Mo., has announced the appointments of **C. H. Mueller** as Vice-President in Charge of Engineering, **J. E. Renner** as Vice-President in Charge of Sales; and **R. E. Crean** as Manager of the company's Philadelphia Industrial Branch Sales Office.

Fred Macks (co-author of "Analysis & Lubrication of Bearings"), formerly head of the bearing research group at NACA, Cleveland, Ohio, and for the past two years self-employed as an Engineering Consultant, has opened a new office at the following address: Willow Lane, Vermilion Lagoons, Vermilion, Ohio.

Inter-State Oil Co., Kansas City, Mo., has announced the appointment of **H. A. Glatte** as Executive Vice-President & General Manager.

Newly elected Officers & Directors of the National Fluid Power Association include: **J. E. Erskine** (Racine Hydraulics & Machinery, Inc.), as President; **O. W. Macy** (Hydraulic Press Mfg. Co.), 1st Vice-President; **J. J. Pippenger** (Double A Products Co.), 2nd Vice-President; with **L. L. Charlson** (Char-Lynn Co.), **F. Flick** (Miller Fluid Power Co.), and **E. M. Greer** (Greer Hydraulics, Inc.), as Directors.

Calumet Refining Co., Chicago, has announced the election of **R. F. Duncan** to the newly-created position of Executive Vice-President.

Douglas Godfrey, formerly with NACA, Lewis Flight Propulsion Laboratory, Cleveland, is now associated with California Research Corp., Richmond 1, Calif. (to conduct research on cer-

(Continued on p. 200)

Deposit Analysis In Lubrication Problems

by P. B. Burgess & J. F. Wygant*

By identification of the phases present in deposits, valuable assistance in solution of lubrication problems may be obtained. A scheme of organic and inorganic phase analysis depending primarily on solvent extraction, X-ray diffraction analysis, and petrographic examination is presented for both routine and investigative purposes. The interpretation of results in terms of operating conditions and causes of lubrication difficulties is discussed.

More often than not, lubrication problems involve deposits either as causes or as side effects. In either case, deposit analysis frequently is invaluable in defining actual operating conditions to indicate causes of lubricant or metal breakdown. This paper presents a scheme for deposit analysis which has been used successfully for both routine and research investigations.

The aims of all deposit analyses are the determination of (1) the sources of deposit components, and (2) the conditions under which deposits are formed. Before these questions can be answered, the lubrication engineer must be informed of the exact nature of the deposit. This is a different concept than mere chemical analysis, and involves determination of the phases present in the deposit. Each phase is a distinct form of matter; i.e., it is a solid, having a certain crystal structure or is noncrystalline, or it is a liquid; and it is a certain chemical element, compound, or solution. It is of utmost importance whether the iron content of a deposit is in the form of metallic iron (α Fe), stainless steel (γ Fe), magnetite (Fe_3O_4), ferric oxide (Fe_2O_3), or rust ($Fe_2O_3 \cdot xH_2O$). For this type of analysis, phase identification methods like organic solvent extraction, X-ray diffraction, and examination with the polarizing petrographic microscope are the principal techniques. Elemental analyses, by wet chemical or spectrographic means, are auxiliary procedures. A variety of other physical and chemical methods, of a diversity limited only by ingenuity, also are useful adjuncts.

ORGANIC ANALYSIS. For most purposes, the organic portion of a deposit can be classified adequately by determination of the water content, solvent extraction of oils, lacquers, and asphaltenes, and ignition of carbonaceous and volatile inorganic materials. More specific information can be obtained by infrared spectroscopy or by other organic analytical methods, but the authors have found such techniques necessary only rarely in routine problems.

Water contents are normally determined by evaporation at 250 F., prior to solvent extraction. Where volatile hydrocarbons or other materials which might be lost with water are present, water content is determined by naptha distillation as in ASTM-D95-46.

Solvent extraction is done in the Soxhlet apparatus using petroleum ether to remove oils; acetone next to remove gums, lacquers, and resins; and finally chloroform to dissolve asphaltenes and alcohol derivatives. After the successive washings, each followed by drying and weighing, a portion of the residue is ignited for estimation of volatile inorganic and carbonaceous materials.

Carbonaceous matter is largely carbon and very-high-carbon residue. Volatile inorganic matter may include some sulfur trioxide from sulfates, water of hydration from hydrated compounds, volatile or unstable halides, etc. In some cases, oxidation may add weight, e.g., to elemental metals or sulfides.

Specific organic tests are sometimes used to identify specific substances. The most common of these are the tests used to determine the presence of anti-freezes which are very undesirable contaminants in motor oils. Both monohydric and polyhydric alcohols can be detected by use of a ceric nitrate

Fig. 1. Petrographic microscope. Solid samples are dispersed in oils of known refractive indices on glass slides to determine the indices of transparent phases by matching. By means of polarizing attachments, crystal geometry may be found as well as size, color, and shape of mineral grains.

*Standard Oil Co. (Ind.), 2400 New York Ave., Whiting, Ind.

(This paper was sponsored by the ASLE Technical Committee on Physical Properties of Lubricants, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

solution. Polyhydric alcohols, and some sugars and other reducible carbohydrates, can be detected by reactions with periodic acid followed by silver nitrate. Sugars can be identified further by water extraction followed by crystallization and identification of the crystals, or by other confirmatory qualitative tests such as reduction of alkaline silver nitrate.

INORGANIC ANALYSIS. The basic methods for the detection of inorganic phases are petrographic examination and X-ray diffraction analysis. Generally, deposit samples are cleaned up by solvent extraction and are examined before ignition. Cleaning can be done by solvent extraction, or by washing and centrifuging. Occasionally, separation of solids can be made on the basis of specific gravity, using heavy liquids. The presence of large amounts of carbonaceous material reduces the sensitivity of the analysis, and it is sometimes helpful to examine residue ashes, keeping in mind the drastic changes in deposit phases due to heating. It is sometimes possible to verify a tentative identification by examination of the ignited sample.

The visual characteristics of deposits should always be noted, since the physical form of the constituents may be of great importance. Metallic iron, for instance, might be in the form of tiny wear particles, filings, turnings, or round shot, depending upon its origin. Lead salts may be in clean discrete particles, or may be embedded in varnish or in hard combustion chamber deposits. The use of a low powered binocular microscope is often helpful. Determination of the magnetic properties of the deposit is also done at this time. Magnetic response may mean that Fe_3O_4 or some other magnetic material, as well as iron, is present, and magnetic separation followed by identification of the magnetic phases is required. This often permits detection of 1% or less of metallic iron, and examination of its form and particle size.

Petrographic examination, Fig. 1, determines the optical properties of the phases present. The degree of opacity, color, crystal form, index of refraction, and other characteristics are determined. Since the optical properties of most minerals are quite distinct, phases often can be identified positively by this means. Since crystals are examined singly rather than in bulk, less than 1% of any one phase may be detected. However, many phases are isotropic and colorless, and some have a wide range of refractive index and other optical properties. These frequently are unidentifiable or can be identified only indirectly. Others may be completely opaque, or glassy and non-crystalline.

Certain amorphous solids and glasses, however, apparently are unique to engine deposits. Familiarity with appropriate phase equilibria thus often permits identification of the phases and estimation of the conditions under which they were formed.

Regardless of optical opacity or color, X-ray diffraction analysis, Fig. 2, permits identification of practically all crystalline phases which are present in amounts of a few percent or more, and of some phases in extremely small amounts. Even phases of the same crystal form almost never have the same basic lattice dimensions. However, the multiplicity of "lines" produced by diffraction from mixtures complicates the analysis so that a small amount of any single phase, say less than 5%, may be missed. The use of petrographic examination improves the sensitivity of this analysis, since often only one or two of the strongest diffraction lines of a phase may be sufficient identification if the possibilities have already been narrowed to a single group of compounds by petrography. The two tools provide a very powerful combination, since they can identify and estimate quantities down to traces of practically all solid materials. Neither, alone, is capable of approaching the effectiveness of the two.

Particle size analyses often are made using the petrographic microscope. This may indicate the source of dirt, such as air-borne dust, or it may be a means of estimating filter efficiency.

Crystallite sizes, determined either by X-ray diffraction or microscopy, may indicate the heat treatment to which deposits have been subjected.

Very thin films of crystalline deposits can be identified by electron diffraction, provided it is possible to mount the tarnished or discolored part in the apparatus.

Additional analytical procedures are used from time to time to provide supplemental information. Important amounts of glassy or other non-crystalline materials must be identified by chemical, or element analysis, methods. Sometimes, the amount of a particular element present in a deposit is itself of significance, as in the case of the metals used in certain oil and fuel additives. The most common of the methods utilized are as follows:

1. Arc and Spark spectrography: for semi-quantitative analysis, especially of the lighter me-

Fig. 2. X-ray diffraction unit. Vertically mounted X-ray source has three windows. Circular cameras are mounted on tracks with the sample held at the center and rocked during exposure. X-ray beams are directed at the sample and diffracted at angles characteristic of the crystal structure of the components. The diffracted beams form lines on the film strip inside around the camera.

tallic elements.

2. Electrolytic solution spectroscopy: for rapid qualitative determination of many metallic elements.

3. X-ray fluorescence spectrography: for quantitative and qualitative analysis of heavier metals (atomic numbers over 22).

4. Conventional wet analysis: used when accurate quantitative data are required.

5. Blow-pipe analysis: for rapid qualitative determination of lead and sulfur.

6. Closed tube ignition: for qualitative detection of water of hydration.

7. Chemical spot tests: for qualitative identification of specific anions and cations, using color reactions principally.

8. Chemical microscopy: similar to spot tests in application, but for very small samples, usually forming identifiable crystalline precipitates; also for some differentiations which are impossible by other quick methods.

9. Differential thermal analysis: used principally when clays or other hydrated compounds are suspected.

ROUTINE PROCEDURE. Because the analytical methods utilized are normally performed by several different people in various parts of the company organization, a well-established routine for sample handling is helpful. The general flow pattern used by the authors is shown in Fig. 3. The key steps in this procedure are (1) the initial examination, upon which is based the choice of analytical methods, and (2) the final evaluation of results. For adequate treatment of deposit analysis, both of these steps should be handled by professionally qualified people.

Another essential to useful routine analyses is an appreciation by all who handle samples that the only worthwhile results are timely results. While accuracy must never be sacrificed, the degree of precision and thoroughness of investigation can vary according to the complexity and importance of individual jobs. Results should be communicated to those requesting the data as soon as they are available. However, interpretations of data should not be made verbally on the spot, but in writing after some period of careful thought. Short, unadorned written reports should follow telephone data as soon as practical for the purposes of confirmation of results, interpretation of findings, and a permanent record to add to the background of future investigations.

For those interested in specific descriptions of the techniques discussed, a bibliography is appended.

INTERPRETATION OF RESULTS. The deposits encountered in lubrication problems are generally derived from:

1. **fuel** combustion and deterioration;
2. **lubricant** deterioration;
3. **machine** wear, abrasion, and corrosion;
4. **manufacturing** and operating contamination.

With some knowledge of the machine and its operation, the sources of deposit constituents can usually be deduced. Furthermore, added information about actual operating conditions is frequently apparent from the makeup of a deposit.

In internal combustion engines, an important source of deposits is fuel, which often contains tetraethyl lead and other additives. Water formed by fuel combustion, under some conditions, enters the crankcase to form emulsion type sludges and, with acidic materials like the oxides of sulfur, to promote corrosion of the engine. Combustion chamber deposits generally contain unburned carbon, with lead halides, sulfate, or oxysulfate appearing at increasingly high operating temperatures. Gum and varnishes from the fuel also may be found in both combustion chamber and crankcase deposits, on intake valves and valve galleys, and on carburetor parts.

Lubricating oils can be oxidized and burned to form resins, asphaltenes, and carbon. With simple acetone extraction, it is not possible to distinguish between the resins and gums formed from fuel and those formed from lubricants. However, asphaltenes, as detected by chloroform extraction, are characteristically a product of lubricating oil deterioration so that some distinction can be made in this way. Besides the organic residues, most modern oils produce inorganic compounds by oxidation and decomposition of various additives. The presence of these compounds is a good indication that deposits are at least partially derived from lubricating oil, provided the type of additive used is known. Oftentimes, the compounds in which additive metals occur vary with temperature, just as in the case of deposits of lead salts.

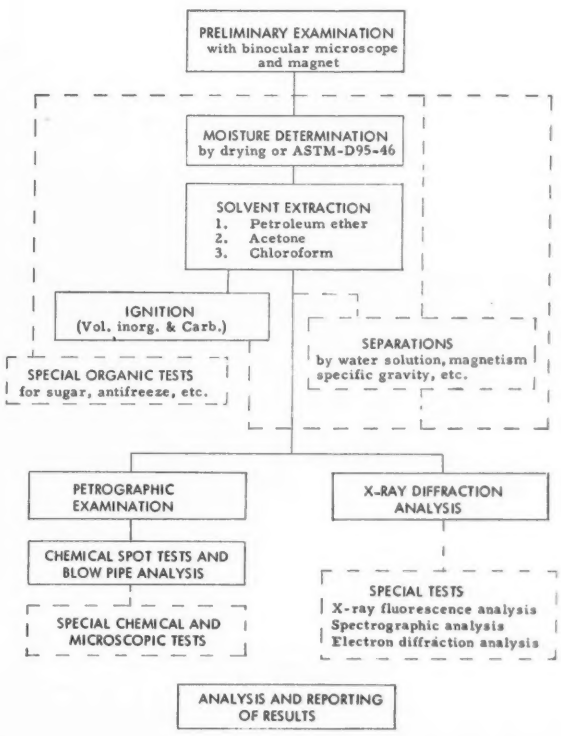

Fig. 3. Routine analytical scheme for deposits.

Corrosion products are always significant when found in deposits. These are sometimes found in situ on the damaged parts, but often are found elsewhere. Perhaps the most common corrosion product is rust. Rust is best defined as hydrated ferric oxide and is characterized by a relatively indefinite structure which makes identification by X-ray diffraction uncertain. However, the appearance is distinctive, and the higher hydrates often are accompanied by identifiable FeO(OH), or Fe_2O_3. The characteristic color and form of rust, as observed petrographically, plus the detection of the water of hydration and of easily soluble iron are used for identification. Confirmation can be obtained by igniting to form magnetite or other crystalline iron oxides. When present in thin layers, rust is insoluble in chloroform, carbon tetrachloride, or other common organic solvents, but dissolves in 10% phosphoric acid in alcohol. The other oxides of iron may be identified readily by X-ray diffraction. Distinctions between the various oxides sometimes permits inference of corrosion mechanisms. However, generalizations without certain knowledge of operating conditions are dangerous since, for instance, magnetite (Fe_3O_4) is normally formed at elevated temperatures, but it also can be formed by aqueous corrosion with a relative lack of oxygen.

Besides oxides, metal sulfides and chlorides sometimes are identified as corrosion products, and may serve as indicators of the corrosive agents. However, many materials serve merely to promote oxidation or aqueous corrosion and do not form characteristic corrosion products. Low melting phosphates, for example, dissolve the normally protective oxide coatings on engine valves to expose them to severe attack by lead compounds. Even small amounts of such materials are of significance, and specific and basic knowledge of the reactions involved is required to determine the cause of difficulty.

Mechanical damage by fatigue, abrasion, or scuffing frequently produces fine metal particles which end up in deposits. The major metals are readily identified by X-ray diffraction and the source of such materials often can be deduced. Aluminum particles in engines, for instance, could come from either pistons or bearings. Iron or steel particles frequently are found as a result of wear of rings, cylinders, and other engine parts; but sometimes, as in the case of fretting damage, the particles generated are so small that they are converted to oxides.

The relative amounts and types of foreign materials in deposits, especially crankcase or fuel system deposits, often pinpoint the source of the deposit. This is true in many cases of contamination by industrial dust, of some by natural dust, and of some due to careless or deliberate introduction of solids.

Although the identification of the components in a deposit can furnish valuable clues to operating conditions, the precise location from which the deposit was taken must be known. This is especially true in internal combustion engines, since wide variations in temperatures produce correspondingly wide differences in deposit make up. The likely presence or absence of water is also important in interpreting results. Every effort should be made, by those who obtain deposit samples, to insure that the sample does, in fact, represent the condition being investigated, and also to see that the samples are adequately identified by source and location. Furnishing all available information on fuels, lubricants, equipment, and history will also facilitate interpretation to secure the maximum benefit from the analytical results.

SUMMARY. Deposit analysis is most useful when the phases present can be identified conclusively. This permits deduction as to the sources and causes of deposit formation, and often results in solutions of lubrication problems. Two techniques of especial value in identifying inorganic phases are X-ray diffraction and petrographic examination. Comparable results for organic phases cannot be conveniently obtained with equipment presently available. However, approximate classification by selective solvent analysis and ignition is generally adequate.

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(Men of Lubrication, from p. 163)

ful and was tested to average load pressures of 4000 psi. In spite of this success it is hard to follow Kingsbury's reasoning on the matter of the central pivot. There are many important thrust bearing problems in which rotation is always in the same direction. The theory of lubrication shows clearly that the pivot must be toward the trailing edge of the bearing for statical equilibrium and he was therefore running a considerable risk of failure in a crucial experiment. The reason the centrally pivoted bearing works is apparently due to a secondary effect of heating and this was not known to him at the time. Nevertheless, the centrally supported shoe is an important practical contribution and is remarkable in that it was achieved simultaneously with the main objective.

Following the successful trial of his invention of the thrust bearing, Kingsbury went to the Worcester Polytechnic Institute in 1899 as professor of applied mechanics where he continued his researches on his bearing. In 1903 he joined the Westinghouse Electric & Manufacturing Co. to work on problems of a mechanical nature. It was here he was finally able to apply his bearing to thrust problems in machinery. His first attempt was not successful apparently because of his too conservative views on operating pressure. He had designed for 50 psi and the bearing ran hot. It was discarded in favor of a ball bearing. In 1909 another opportunity was afforded him to apply the bearing to a large vertical motor. In spite of its complete success the company officials would not approve its adoption and the motor was equipped with a roller bearing before it was delivered. Kingsbury seems to have got a good deal of satisfaction out of being asked many years later to replace the roller bearing on this same motor with one of his own bearings.

Kingsbury obtained a U. S. patent on his thrust bearing in 1910 though not without trouble. His first application (1907) was denied because of Michell's British patent of 1904. But Kingsbury was able to establish his precedence in the matter. Thereafter his bearing soon gained acceptance. The Westinghouse Machine Company, and later the Westinghouse Electric Company, took licenses to build the bearings.

For some years Kingsbury sold bearings which were built to his specifications by the Westinghouse Machine Company, but in 1921 he seems to have decided to do all his own manufacturing and his company, the Kingsbury Machine Works, is today a very successful organization.

Albert Kingsbury died in 1943 after a distinguished career in engineering and business. His invention of the thrust bearing was a great technical

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achievement and eventually led to financial success. But Kingsbury did not become a mere man of business. The scientific side of his company's activities was much in his mind and the research produced by Kingsbury and those who worked with him is distinguished by its technical excellence and evident care in experimental execution. Kingsbury's electric analogy to the problem of hydrodynamic lubrication is in itself sufficient to have given his name a significant place in the development of lubrication science.

Albert Kingsbury was not a narrow man of engineering science. His father was said to have been an expert musician, and as a young man the son played the flute and sang in a church choir. In his later years he devoted much time to the cultural pleasures to be derived from reading in the original language the literature of Italy, France, Germany, Spain and others. He was a man of the highest character, and this is perhaps no where so well illustrated as by his refusal to advertise his company's product, apparently on ethical principles.

Albert Kingsbury received many honors during his lifetime from learned and engineering societies and others for his accomplishments. In 1950 the University of New Hampshire named their new College of Technology "Kingsbury Hall" in honor of Albert Kingsbury, "First Professor of Mechanical Engineering at the University of New Hampshire . . . A Great and Honorable Man."

*This model and the torsion compression machine are both preserved by the University of New Hampshire in their Kingsbury Hall.

ACKNOWLEDGMENTS. The main sources consulted in the preparation of this article are: Kingsbury's autobiographical account, "Development of the Kingsbury Thrust Bearing," Mechanical Engineering, December 1950; and a booklet on the dedication of Kingsbury Hall at University of New Hampshire, October 1950. The author wishes to thank Mr. Nelson Ogden of The Kingsbury Machine Works for help during the preparation of this article.

(Lube Lines, from p. 149)

with the job" that was most stimulating to this observer and which was also remarked by others. With this as the prevailing attitude it was not surprising to find that something concrete had been accomplished by nearly every committee. Samples:

The job done by the Program Committee sets the tone of the whole Society. Their performance in assembling some fifty papers of first rate quality from varied sources has been as outstanding as it must have been difficult.

A new Industry Advisory Committee composed of industrial member representatives came up with a recommendation for circulation of annual committee reports as a means for getting people better acquainted with what we are doing.

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The Principles & Practices Committee has developed a tentative definition of a Lubrication Engineer, and has taken on the job of preparing a basic outline of the steps involved in establishing a lubrication engineering service in shops where none now exists. Talk about ambition!!!

For the first time, the Editorial Committee had copies on hand of all the papers presented at the convention. That's a lot of meat to serve up in future issues of "Lubrication Engineering."

The Handbook Advisory Committee has made a good start on the outline of the proposed Handbook, after getting confirmation of the editors and an agreement with the publisher.

A new Self Survey Committee has been appointed to take a long look at ASLE operations, activities, etc. It is expected that their work will be considerably facilitated (?) by the previous experience of the U. S. Government's Hoover Commission in a like task.

These are only a few which are cited to indicate a new ASLE atmosphere. More will doubtless be forthcoming as other comments come in. (H. E. Mahncke)

(Section News, from p. 173)

presented a paper entitled "Design, Lubrication and Failure Patterns of High Speed Anti-Frictional Bearings," in which emphasis was placed on high speed and exceedingly high speed service of bearings. 'Whereas spindle speeds of 5,000 rpm are considered high now, the coming spindle speeds will be 20,000 to 25,000 rpm; temperature will go up from the present common range of 300 F. to 900 F.' (Submitted by T. R. Costello, Sec'y-Treas.)

LOUISVILLE, February. E. C. Carmichael, Socony-Vacuum Oil Co., presented a paper entitled "Fundamentals of Compounding and Selection of Lubricants."

MILWAUKEE, March. J. T. Wilson, Louis Allis Co., presented a paper entitled "Industrial Significance of the Incompatability of Greases and the Effects of Industrial Contamination."

April. L. R. Tharp, E. F. Houghton & Co., spoke on the subject "Drawing Compounds, Their Functions, Make-Up & Selection." (Submitted by J. A. Gramling, Sec'y.)

NEW YORK, March. C. M. Larson, Chief Consulting Engineer of Sinclair Refining Co., presented a paper entitled "Additives for Fuels & Lubricants." (Submitted by O. W. Wuerz, Program & Vice-Chrmn.)

N. CALIFORNIA, March. Visit to the United Air Lines Maintenance Base at the San Francisco Airport, with R. D. Kelley, Superintendent of Technical Development, acting as host. Mr. Kel-

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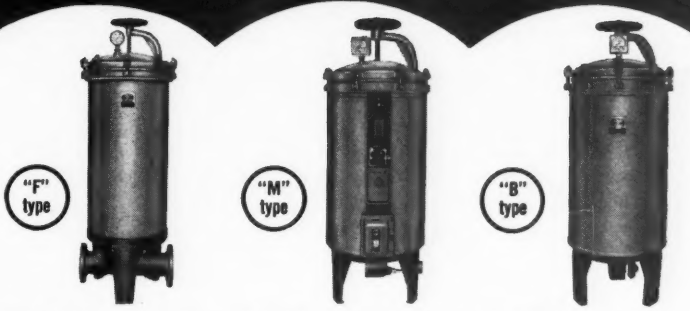

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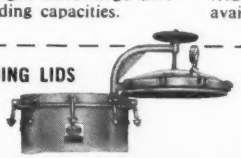

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INDIANAPOLIS, IND.

ley compared the company's first plane, the Boeing 80A tri-motored plane, with the Boeing 707 jet plane expected in four or five years, and pointed out that although the jet plane will provide as much transportation as 49 of the older type Boeing planes, the '707' will use only one-half as much fuel in doing so.

April. Election of Officers (see ASLE Directory), followed by an address by W. C. Bauer, Bowser Inc., entitled "The Maintenance in Service of Lubricating Oil Quality by Filtering, Centrifuging & Reclamation," in which the various means currently in use to maintain the quality of oil in operating machinery were discussed. 'Rather than following the practice of draining the lubricating oil at frequent intervals as is commonly recommended by oil companies, the oil may be continuously maintained in a clean and pure condition — a process referred to as rectification. Thereby the service life of the oil may be extended and lubricant cost minimized. In long continued use, mineral lubricating oils do not wear out, they do not lose their inherent value as satisfactory lubricants, nor do they change in essential characteristics. Thus, the maintenance of satisfactory lubricating oil quality in service resolves itself to the removal of contaminants—both solid particles and dissolved materials — from the used oil. The oil companies have sought to provide a lubricant that would perform its desired functions for as long as possible without any rectification.' Mr. Bauer suggests that the oil companies should provide a lubricant that would lend itself to complete rectification more easily.

May. S. D. Craine, Service Engineer & Consultant for Jones Foundry & Machine Co., presented a paper entitled "Preventive Maintenance of Industrial Gears," in which he discussed a planned preventive maintenance program and its importance in the operation of industrial gears. 'A gear preventive maintenance program requires the training of a person to, first, recognize the external symptoms of improper gear operation; second, inspect the gears and interpret the early indications of gear difficulties; and third, take action to correct the condition. The plant lubrication engineer is the logical person to handle this program since he or his staff comes in frequent observation of the gears. He must, however, be able to recognize not only difficulties due to improper lubrication but also due to all other causes.' (Submitted by G. H. Hommer, Reporter, & A. C. West, Sec'y.)

TWIN CITIES, March. Dr. P. A. Asseff, Lubrizol Corp., presented a paper entitled "Anti-Wear Requirements of Modern Motor Oils." (Submitted by C. D. Johnson, Sec'y.)

WHEELING, April. A. Kommer, Ross Enterprises (distributors of Briggs filtration equipment), presented a paper entitled "Oil Filtration." (Submitted by S. J. Litten, Sec'y-Treas.)

(Personals, from p. 192)
tain fundamental aspects of lubrication and wear).

C. A. Norgren Co., Englewood, Colo., has announced the appointment of **W. O. Hall** as Chief Field Engineer.

Newly elected Officers & Directors of the Internal Combustion Engine Institute include: **B. G. VanZee** (Minneapolis-Moline Co.), as President; **E. D. Tull** (Cummins Engine Co., Inc.), Vice-President; **P. Norton** (Wisconsin Motor Corp.), Treasurer; **H. W. Smith** (Caterpillar Tractor Co.), Secretary; with former president of the Institute, **J. E. Heuser** (Westinghouse Air Brake Co.), as a Director.

Climax Molybdenum Co., NYC, has announced the appointment of **L. E. Ekholm** as Manager of Sales.

Lecturers for the Lubrication Engineering Course (ASLE 10th Annual Meeting, Chicago) included: **C. A. Bailey** (U. S. Steel Corp.), **Dr. W. E. Campbell** (Brush Labs. Div., Clevite Corp.), **S. D. Craine** (W. A. Jones Fdry. & Mach. Co.), **W. T. Everitt** (Eastman Kodak Co.), **Dr. D. D. Fuller** (Columbia University), **J.**

E. Gleber & K. S. Smiley (E. I. duPont deNemours & Co., Inc.), **A. J. Gruenewald** (Lincoln Engineering Co.), **T. M. Murphy** (Alemite Div., Stewart-Warner Corp.), **K. E. Paulus** (Bowser, Inc.), **C. E. Schmitz** (Crane Packing Co.), **W. M. Schuck** (Armco Steel Co.), and **R. J. Torrens** (Eastman Kodak Co.).

Obituaries: **W. A. Fletcher** (ASLE N. California Section), E. F. Houghton & Co., San Francisco, April 4; **F. A. Hiter,** Senior Vice-President, Stewart-Warner Corp., Chicago, April 9.

Lubrication in the News

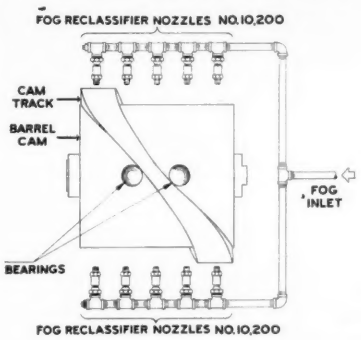
Mount Palomar's 'Horseshoe.' Plastic models of huge movable parts of powerful generators and bearings are the subject of concentrated study at the Research Laboratories of Westinghouse Electric Corp. Among the demonstration models is this replica of one of the two principal bearings, built by Westinghouse, which support the mammoth 200-inch telescope atop Mt. Palomar, near Pasadena, Calif. Research engineer Howard N. Kaufman here moves the "horseshoe" with ease as oil is forced into the pads — or receptacles — from the reservoir beneath them. Without the oil, friction between the two surfaces, both on the model and on

Norgren

MICRO-FOG LUBRICATION

- **eliminated flaking and galling of cam track**
- **eliminated heating and excessive wearing of bearings**
- **eliminated constant squealing of barrel cam**
- **eliminated daily manual lubrication**

ON 6 PRINTING PRESSES
For UNIVERSAL MATCH CORP.



Oscillating movement of table on the printing press is achieved by a 16" dia. barrel cam rotating at 150 rpm. A Norgren Lubro-Control Unit, Model 3765-2 (Filter, Regulator, Lubricator) was installed to lubricate the cam, cam track and follower roll bearings. It solved the lubrication problem so successfully that similar units were installed on five other presses of the same type.

Two ¼" pipe lines, 18" long, carry Micro-Fog to 5 reclassifiers on each side of barrel cam. 10 psi air pressure used. Solenoid valve coordinates lubricant flow with press operation.

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- Saves on Repair Bills
- Eliminates Costly Hand Oiling
- Reduces Downtime
- Lengthens Machine Life

WITH Bijur Automatic Lubrication as an integral part of your equipment, your customers benefit from continuous peak production. Plant after plant reports increased out-put and greatly reduced maintenance costs with Bijur-equipped machines. Costly downtime for lubricating by hand is eliminated. Every bearing is metered the proper quantity of oil at predetermined intervals. Work spoilage and bearing troubles caused by over-lubrication are avoided. Fire hazards are reduced. Personnel accidents are prevented.

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More than a million Bijur-protected machines are in use. Hundreds of leading manufacturers standardize on Bijur as "built-in" components of their machines. Bijur emphasizes custom-engineering, and we will gladly cooperate in designing a system to meet the specific requirements of your machine.

Design Bijur into your machines now in production or in the planning stage. Write for literature and engineering information.

0-3

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Rochelle Park, New Jersey

the telescope, would cause the bearing to stick firmly. Westinghouse engineers designed the flotation method so that with a film of oil, three-thousandths of an inch thick, the huge 500-ton telescope could be moved with a ½-horsepower motor — only slightly larger than the motor in a washing machine. This oil flotation process is the same as is sometimes used in construction of waterwheel generators.

Book Reviews

Physical Properties Of Lubricants, by D. D. Fuller, et al, American Society of Lubrication Engineers, Chicago, 1951, 69 pages, price $1.00.

The properties of lubricating oils can be characterized by a series of physical and chemical tests. This book collects the twelve most important properties into a readily available form. Each property is defined in terms that are easily understood, and the test methods used to determine the property are given. The significance of these tests is given although their reproducibility is not. As might be expected, viscosity is treated in greatest detail. Numerous viscometers are described. The general uses and common classifications of lubricating oils are included. The effects of temperature and pressure on viscosity are covered. Other properties described in the book are density and specific gravity, cloud and pour points, flash and fire points, carbon residue, neutralization number and interfacial tension, saponification number, emulsification and specific heat. One property which might have been included but which is not discussed is the shear stability of high V.I., polymer-containing oils under severe working. An extensive bibliography and index conclude the book.

(Reviewed by K. H. Strauss, The Texas Co.)

(New Products, from p. 186)

touching either the hook or barrel), heavy barrels stored in contact with each other can be easily loaded on the truck by a single operator; special design of the loading shoes provides easy tipping of the heaviest barrels. These same features make it possible to unload barrels for storage without space between by merely releasing the barrel at the desired location, as barrel slips off shoe when cart is tipped forward. (Valley Craft Products, Inc., LE-11/3, 760 Jefferson Ave., Lake City, Minn.)

Centro-Luber System. A simple, positive, high pressure centralized system has been announced for controlled lubrication of bearings while machine is operating or while idle. The pumping unit is manually-operated and incorporates a 15 lb. capacity reservoir for grease or oil. A few strokes of the pump handle forces lubricant under 2500 psi pressure through a single line circuit of injectors, or adjustable hydraulic force-feed measuring valves, one for each bearing. These injectors, or valves, in turn deliver a measured quantity of oil or grease to each bearing every time the system is cycled. After a few strokes of the pump handle, an indicator device shows when lubrication cycle is complete. Release of pump handle by operator automatically vents and recharges system. Pump has button head filler fitting for refilling reservoir from a bucket pump, or may be hand-packed. System comes complete with all necessary accessories; no special tools or engineering knowledge required to install. (Bulletin 807, Lincoln Engineering Co., LE-11/3, 5743 Natural Bridge Ave., St. Louis 20, Mo.)

Non-Oily Penetrant. Savings of time and equipment are promised by a new product, the first non-oily penetrant for loosening corroded nuts or bolts and larger sized parts of mechanical equipment. An odorless chemical formulation, the penetrant is fast acting, penetrates deeply, is non-flammable, and does not spread over the work. It can be applied by pour spout or squirt gun; after several minutes the work is tapped with a hammer and parts can then be disassembled. (Olin Mathieson Chemical Corp., LE-11/3, 10 Light St., Baltimore 3, Md.)

Full Flow Filters. A new line of full flow filters offering flow rates up to 900 GPM has been introduced that can be used for filtering a wide variety of liquids including engine lube and fuel oils, metalworking oils and coolants, and parts washing solutions. Designed for in-line installation, the new filters combine compactness with exceptionally high flow capacity to insure continuous filtration of entire oil or coolant supply. Nine different sizes are available using one to 36 cartridges. Extra-large dirt holding capacity is obtained by new "Flo-Pac" cartridge designed especially for this filter. Resin-impregnated pleated paper cartridge has 44 square feet of filtering surface. Filter element in cartridge is completely inert and will not remove additives found in many oils. It will not soften or disintegrate when used on coolants or other liquids having a water base. All models feature quick opening covers for convenience in changing cartridges. Covers are fastened by swing bolts which one man can loosen quickly. Large models have unique lifting device that swings cover to one side for easy access to cartridge. A permanent neoprene gasket seals cover tightly to filter tank. Cartridges are sealed by a removable hold down cap; spring on cap compresses when cover is closed to prevent bypassing. (Houdaille-Hershey of Indiana, Inc., LE-11/3, 20 South Ave., Lebanon, Ind.)

Parker Toggle Clamp. Model 824 tube bender, a bench-mounted manual bender widely used in fabrication of metal tubing for fluid-handling systems, is now available with a new toggle clamp for easier and speedier operation. The toggle clamp, through simple handle action, saves time by providing quicker gripping of the tube to be bent. The clamp is at-

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(polyalkylene glycols and derivatives)
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FOR **POWER LUBRICATION**

Applications Unlimited . . .

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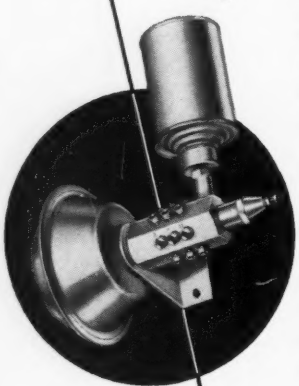
Adopted by leading Fleet and Bus Operators for automatic, controlled lubricant application each time driver applies the air brake. Complete Systems available in handy kits for simple installation. Air-Operated Multi-Luber Systems may be used wherever compressed air is available to reduce operating costs and increase efficiency on equipment ranging from transport trailers to automated, high speed lathes.

Where compressed air is not available, the air cylinder of the Multi-Luber can be replaced with a push button. These Manual Systems are available in kits for quick installation on tractors, farm implements, and a wide range of industrial machinery.



VACUUM-OPERATED

Multi-Luber Systems are also available for instantaneous, automatic lubrication of equipment ranging from light trucks to fork lift trucks, or for any application where vacuum is available. A touch of the control button, located wherever desired, delivers a pre-measured quantity of refinery-pure lubricant.

AND NOW . . . AVAILABLE ON 1955 model LINCOLN and MERCURY motor cars

Here is the newest and most revolutionary application of Lincoln's vacuum-operated Multi-Luber System. Now, purchasers of new Lincoln or Mercury motor cars have available instantaneous Power Lubrication at their own convenience. A mere touch of a button on the instrument panel provides the continuous pleasure of smoother car performance, greater steering ease and increased operating economy.

*Trade Name Registered Patent Pending

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PIONEER BUILDERS OF LUBRICATING EQUIPMENT FOR A QUARTER CENTURY

tached merely by inserting its swivel block pin into the third hole of the bender's radius block. The toggle clamp, without shoe insert, accommodates tubing of 1½" outside diameter. Shoe inserts are available for tubing sizes ranging from ⅜" through 1¼" outside diameter. (Catalog 1141A10, Parker Appliance Co., LE-11/3, 17325 Euclid Ave., Cleveland 12, Ohio.)

'Contact,' specially formulated for the electronic and electro-mechanical industries, is a new colloidal solution with a hydro-carbon carrier base. Super-capillary action forces it into inaccessible places where it instantly cleans and deposits a coating of 'duralube' — a hard-bonded dry dubricant which resists corrosives, heat, and cold. Nonflammable and completely harmless to precious metals and all contacts, 'Contact' leaves no gummy deposit and does not effect resistance, capacitance, or inductance. (Beaver Laboratories Inc., LE-11/3, 86-51 Palo Alto Ave., Hollis 23, N. Y.)

Ucon Coating Fluids. A new series of rust-preventive fluids, for coating metal parts before storage, can be burned cleanly from metal surfaces after storage. Marketed under the brand name 'Ucon,' they do not leave residues after burning to interfere with painting, welding, soldering, brazing, or annealing. (They can also be removed with solvents or, in some cases, by mild scouring with water.) Available in several viscosity grades, they can be used diluted or undiluted; choice of method depends on the thickness of the residual film necessary to provide complete protection on the particular metal, facilities in the storage area for accommodating solvent vapors, and the economics of using inexpensive solvents as diluents. (Carbide & Carbon Co., LE-11/3, 30 E. 42nd St., NYC 17.)

Multiple Branch Tee. A 'Tube-Turn' multiple branch tee with integral extruded outlets has been introduced that makes possible a sound and economical solution to

many header problems in the petroleum, chemical, and power fields. Conforming to the Code for Pressure Piping area replacement formula, it is especially useful where branch outlets are appreciably smaller than the run pipe. Tailored to individual jobs its size, wall thickness, and the number and size of the outlets are determined by individual service conditions. (National Cylinder Gas Co., LE-11/3, 224 E. Broadway, Louisville 1, Ky.)

Grease Penetration Cone. Made of magnesium, with stainless steel tip, and weighing 102.5 grams, a new grease penetration cone (developed by Standard Oil Co. (Ind.) Research Laboratories) has been made available that will run all grease tests where present ASTM brass grease cone is used, plus additional tests requiring penetration up to 61.5 m.m. in semi-fluid greases. (Bulletin 4109, Labline Inc., LE-11/3, 217 Desplaines, Chicago, Ill.)

(Current Literature, from p. 177)

truck front wheel bearings. Lubrication of wheel bearings on cars equipped with disc brakes is also covered. (National Lubricating Grease Institute, LE-11/3, 4638 J. C. Nichols Pkway, Kansas City 12, Mo.)

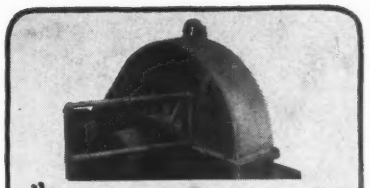
"Torrington Maintenance Manual." A practical check list of 19 steps for proper maintenance of rolling mill bearings, and comprehensive information on installation and use, are presented in this new illustrated manual. Covering mountings, installation, lubricating and fitting practices, special and detailed attention is given to organizing, equipping and training the selected personnel of maintenance departments. Supplementary information explains individual considerations affecting roll neck bearing service life and tells how to prolong it. Elimination of misalignment and avoidable bearing wear are characterized as jobs in which engineers must cooperate from design to maintenance. Major causes of misalignment are listed as foundation settlement, housing distortion, unsatisfactory lubrication, differential heat expansion, insufficient structure strength, improper installation, and inadequate maintenance. The manual comments that the factor of maintenance is particularly important in rolling mills because perfect alignment is impermanent and only preventive and corrective bearing maintenance schedules can enable the mills to produce with top economy. (The Torrington Co., Bantam Bearings Div., LE-11/3, South Bend, Ind.)

Field Report No. 140, a "how-to" discussion on the lubrication of engines and compressors, has been made available that highlights excerpts from an article entitled "New Methods for Engine & Compressor Maintenance," by R. S. Ridgeway, Standard Oil Co. of California ("Petroleum Refiner," Jan. '54). (The Alpha Corp., LE-11/3, 65 Harvard Ave., Stamford, Conn.)

"Automatic Loading Rack Flash Point Tester," Bulletin 720, points up the safety and time-saving features of the apparatus when used as a means of checking distillate fuel at any bulk distributing or storage point. (Precision Scientific Co., LE-11/3, 3737 W. Cortland St., Chicago 47, Ill.)

'54 Supplements to Book of ASTM Standards, Part 5, "Fuels, Petroleum, Aromatic Hydrocarbons, Engine Antifreezes" (308 pp.), includes 36 standards covering crude petroleum; butadiene; motor and aviation fuels; petroleum solvents and napthas; kerosene and illuminating oils; lubricating, turbine, and electrical insulating oils; plant spray oils and petroleum sulfonates; lubricating greases; paraffin and hydrocarbon waxes, gaseous fuels; engine antifreezes; thermometers; hydrometers; and general testing methods. (American Society for Testing Materials, LE-11/3, 1916 Race St., Philadelphia 3, Pa.; price $3.50.)

Field Report No. 141, a test report on the use of Molykote lubricant for highly loaded threaded con-



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"The herring-bone gears in the drive unit of our 6-stand, 10-inch mill that rolls our high speed tool steels became noisy. Inspection showed definite signs of flaking of gears. This was in 1939. It was then we started to use LUBRIPLATE in them and we have not encountered any flaking trouble since."

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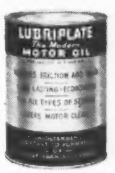

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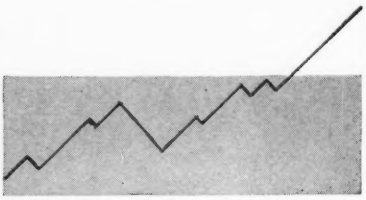

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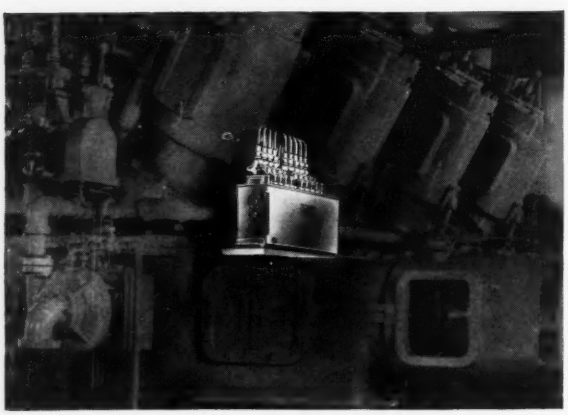
Manzel Force Feed Lubricators insure efficient machinery operation by lubricating automatically. No stops for hand oiling — or because of breakdowns caused by faulty lubrication. They



quickly pay for themselves through savings in down time, labor and lubricants.

These sturdy, dependable lubricators can be installed on new or existing equipment. Write for details.

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nections, describes in detail comparison tests between Molykote Type G (a highly concentrated MoS_2 paste) and other thread lubricants at room and elevated temperatures. Tests covered: the effect of the Molykote on bolt tension; the effect of galling in the threads and below the nut; the effect of repeated tightening on bolt tension; and determination of the coefficient of friction. Several curves illustrate the results obtained. (The Alpha Corp., LE-11/3, 65 Harvard Ave., Stamford, Conn.)

Hydraulic Standards Available. A revised up-to-date issue of the JIC Hydraulic Standards for Industrial Equipment has been made available, presenting the Standards in complete, unabridged form and including all revisions made at the January '53 meetings of the Joint Industry Conference. Also included are the 'Recommended Practices on Hydraulic Packings & Seals,' examples of packing code identification, sample circuit using JIC symbols, glossary of terms, and other useful information. (Miller Fluid Power Co., LE-11/3, 2040 N. Hawthorne Ave., Melrose Park, Ill.)

Pipe Line Filters Detailed. Type PL pipe line filters, designed for removal of water, oil, dirt, and pipe scale from compressed air or gas lines, are detailed in a new brochure containing engineering, operating, and application information. The filters have been engineered for pressures up to 150 lbs.; standard models are furnished for pipe sizes to 2″ while special models are available to 6″ pipe sizes. Design features include extremely low pressure drop which does not exceed three ounces; ease of disassembly for inspection purposes; permanent type separator-element which may be easily cleaned in hot water; an element of brass and bronze construction; corrosion-proof steel shell; cast-iron head, threaded inlet and outlet connections; an absence of moving parts; hydrostatically pressure tested to 250 lbs. psi. (Air-Maze Corp., LE-11/3, 25000 Miles Rd., Cleveland 28, Ohio.)

E. S. P. S., Inc.

The following Positions-and-Engineers Available information is furnished by **Engineering Societies Personnel Service, Inc.,** a non-profit placement service sponsored by **ASLE,** AIEE, AIME, ASCE, ASME, ESD, ECSF, ESM, ISPE, SNAME, and WSE. Replies should be addressed to the key numbers indicated.

If placed in a position as a result of these listings, applicant agrees to pay an established moderate placement fee which is used to defray the expenses of this non-profit, self-supporting service (rates available upon request).

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ENGINEERS AVAILABLE

Chemist (194-LE) 41, Ch. Eng.; 11 yrs. Industrial finishes covering lacquers, varnishes, paints, & allied products; 3 yrs. formulating petroleum colloidal products including plastics, rubber, & allied industries; $6,300; Midwest.

Ch. Engr. (195-LE) 50, BSME; 16 yrs. design, production & sales engrg. for bearing mfr.; 8 yrs. design, chemical research, & sales of petroleum eqpt.; 4 yrs. supervise operation of bldg. materials mfg.; $10,000; US.

Geologist (196-LE) 34, MS. Geol.; 7 yrs. geological & mining engrg.; 15 mos. drafting for steel corporation; 1 yr. petroleum geologist making microscopic studies on well cuttings, well log compilation, & well scouting; $6,000; Midwest.

Sales Engineer (197-LE) 33, Ch. E.; 6 yrs. technical serv. on distillation & cracking eqpt., acid & oil manufacturing; quality control & improvement of manuf. methods; $6,000; Chicago.

Purchasing Agt. (198-LE) 31; 5 yrs. buyer of pipeline material, establish procedures, editing purchasing manuals, surplus sales, supv. expediting & clerical force, and inspect manuf. facilities; $7,000; US.

Purchasing Agt. (199-LE) 44; 12 yrs. purchasing general mills supplies including steels, forgings, castings, ferrous & non-ferrous metals, hardware, piping, plumbing & eqpt. for constr. of fractionating tower pressure vessels, etc.; $5,200; Chicago.

Sales Engr. (200-LE) 29, ME; 6 yrs. selling industrial clutches, torque converters, oil seals & "O" rings, and general machinery; $7,200; Wisconsin.

Geodetic Engr. (201-LE) 36, CE; 2½ yrs. field measurement & office computation of all stream flow data; 8 yrs. supv. of seismographic surveying & computation, locate concession corners by astronomic means for oil company; $6,000; West.

Appraisal (202-LE) 44, ME; 10 yrs. design & constr. of natural gas & liquified petroleum gas standby services for industry; 2 yrs. supervise, plans & specs. for power plants; 2 yrs. supv. operations of compressor stations; $8,400; Midwest.

Ch. Engr. (203-LE) 33, ME; 3 yrs. organization, operation planning, design of jigs & fixtures, coordination & development of production loading & assembly techniques of research designed rockets; 1½ yrs. consultation on mech., chem., & production engrg. problems; $12,000; Midwest.

Gen'l. Mgr. (204-LE) 37, ME; 5 yrs. chief engineer for consultant; 3 yrs. develop new models of combustion type heaters, custom eqpt. for central system oil clarification & purification, and resp. for design & calculations, drafting, model making & testing; 1 yr. assist in the design of coal burning gas; $12,800; Midwest.

Geologist (205-LE) 30, Geol.; 28 mos. report writing, repr. on exploratory wells, maintaining watch on competitor exploratory activity, maintaining up-to-date interpretation of sub-surface geological structure; $4,200; Midwest.

Chem. Engr. (206-LE) 32, MS; Chem. Engr.; 32 mos. study of polymerization & wax de-oiling processes; 42 mos. heat transfer study of high temperature gasification of hydrocarbons; 34 mos. operation of pilot plant units for study of hydrogenation, alkali refining & deodorization of edible oils; $6,500; Chicago.

Sales Engr. (207-LE) 55, EE; 8 yrs. educating distributor & salesman of special lubricants; 2 yrs. contacting yards & quarries for bldg. matl. mfgr.; 1½ yrs. design of high tension power lines; 2 yrs. supv. all dept. production & shipments; $7,200; Chicago.

Process Engr. (208-LE) 40, Ch. Eng.; 2½ yrs. technical service, plant problems, combustion engr. on oil heaters; 3 yrs. traveled to refinery eqpt. suppliers checking specs. of steel plate, pipe, flow meters, pressure vessels, pumps & misc. eqpt.; $500; Midwest.

Petr. Engr. (209-LE) 32, ME; 8 yrs. in petroleum industry handling construction project work; design, purchasing, costs & sub-contracts; $7,800; US.

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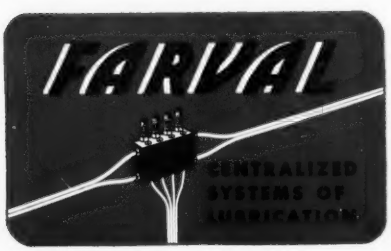

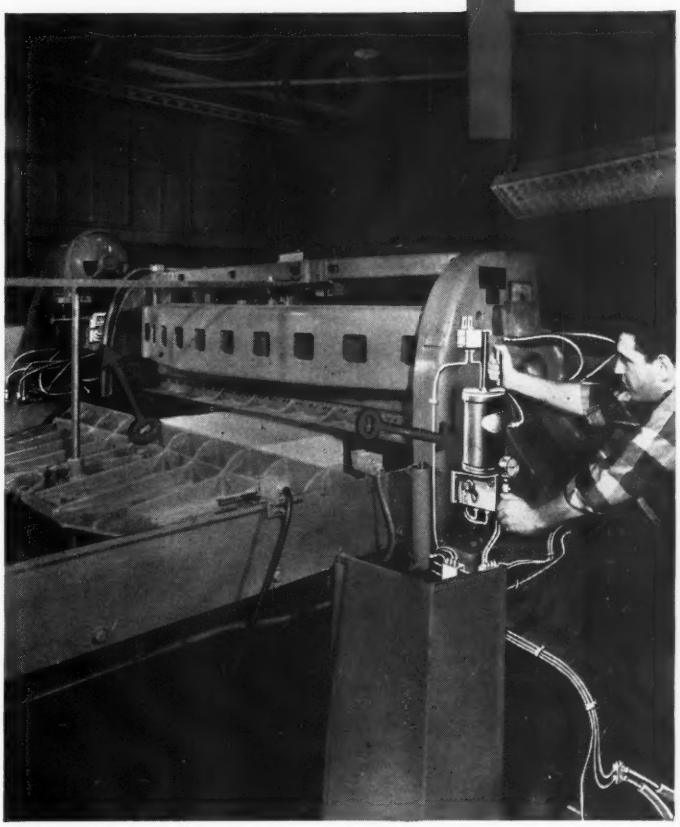

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